COMPUTER AIDED DESIGN OF BUTTERFLY VALVE

by

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CERTIFICATE

29.2.96

It is certified that the work contained in the thesis entitled " **Computer Aided Design Of Butterfly Valves"** by *Sqn Ldr. Brijendra Singh* has been carried out under my supervision and it has not been submitted elsewhere for a degree.

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BRIJENDRA SINGH
(Sqn.Ldr)

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LIST OF SYMBOLS

a	Moment arm (mm)
Ac	Bolt area (sq mm)
b	Width of key (mm)
d	Valve shaft diameter (mm)
da	Flange outer diameter (mm)
D	Diameter of butterfly valve disc
d_{EO}	Equivalent orifice diameter (mm)
Da	Body wall outer diameter (mm)
dc	Core diameter of bolt (mm)
Di	Body wall inner diameter (mm)
d1	Inner diameter of split thrust ring (mm)
d3	Outer diameter of split thrust ring (mm)
di	Flange inner diameter (mm)
dt	Bolt circle diameter (mm)
dL	Pipe outer diameter (mm)
dD	Diameter at the location of gasket load reaction (mm)
f	Friction factor
H_L	Head loss (mm)
h_F	Flange thickness (mm)
h_A	Length of flange through hub (mm)
K	Head loss factor
Kv	Butterfly valve loss coefficient
Kr	Bearing torque coefficient
L	Length of key (mm)
Mb	Bending moment (N-m)

M_T	Twisting moment (N-m)
Mv	Resultant moment (N-m)
n	Numbers of key
N	Numbers of bolt
P	Pressure (bar)
P_{RZ}	Additional pipe forces for unspecified conditions
R	Radius of curvature of disc (mm)
Sr	Body wall thickness (mm)
S	Design factor of safety
S_F	Thickness of hub at the back of flange (mm)
t1	Depth of shaft key (mm)
V	Average velocity through restrictions (m / sec)
μ	Coefficient of friction
σ_{zul}	Allowable design stress (MPa)
σ_{vorth}	Stress developed (MPa)
σ_1	Calculated stress in flange through section A-A (MPa)
σ_2	Calculated stress in flange through section B-B (MPa)
σ_3	Calculated stress in flange through section C-C (MPa)

ABSTRACT

In the present Work, a methodology for interactive design of BUTTERFLY VALVE has been developed. Based on the proposed methodology, a software package for interactive design has also been developed.

In this thesis, specific problem area encountered in the design of butterfly valves, and fluid mechanics for designing valve has been discussed .

The overall design procedure covers the design of major components of butterfly valve and then carry out stress analysis for safety of design. Extensive use of C-Graphics has been used to display sectional-view each component after design which makes the package more realistic. Finally the program draws the assembled view of all the designed components of the Butterfly valve. Here both I.S and B.S codes has been used for selection of materials for parts. The package can be use as an educational aid as well as by the Industries for the design of Butterfly valve.

CHAPTER-1

INTRODUCTION

1.1 HISTORY

When man first used bamboo for pipelines, about 4000 years ago, a method to start and stop flow was first encountered, and the first plug valve was invented. This event was said to have taken place in China. Artifacts from the depth of mediterranean contained fragments of petcocks dating back to before the time of Christ. During the time of Roman empire, wooden valves were used that looked much like the valves manufactured today for use in wine casks and beer kegs.

Nearly every person in this civilized world now comes into contact with valves. They are the main controlling elements in the fluid-handling systems. Valves serve five primary functions.

1. To start and stop the flow.
2. To regulate and throttle the fluid flow.
3. To prevent back flow.
4. To regulate pressure.
5. To relieve excessive pressure.

How well these functions are accomplished largely determines the performance of the system.

Management is becoming more and more aware of the importance of valves in the industrial plants and processes. In the hydrocarbon and natural gas industries, valves represent about 10% of the new plant capital expenditures and 15% of the maintenance budget for replacement purchases.

Until the late 1950s, valve manufacturers generally kept pace with industrial and military demands. Space age technology then forced them to meet strange and unforeseen specifications. Fluid system control became a major problem in the design and development of missiles, advanced aircrafts, hypersonic testing facilities, and space vechicles. Enginneers were called on to design valves that could control extremely cold or hot, noxious, highly reactive, intractable,

self igniting fluid, valves that could operate at both high and low temperatures and pressures and high vibration levels, and could be lightweight and remotely operated.

1.2 Specific Problem Areas

Leakage of expensive, toxic, corrosive, or explosive fluids cannot be tolerated. Great effort is being expended to attain the nearly impossible goal of a zero leak valve.

The material selection problems facing industrial valve designer are extremely difficult. The following factors must be considered .

Large temperature excursions : The anealling effect of temperature cycling can change physical properties. Differential theramal expansion can cause warping and bending.

Vacuum : Many materials outgas in a vacuum environment and undergo property variations.

Extreme temperatures : Material looses strength at very high temperatures, and welding can occur. At low temperatures metals become brittle and sealing materials lose plasticity.

Compatibility : Some material, for example react with liquid oxygen when shocked. Metal parts may gall and experience excessive wear.

Radiation : Some nuclear valves must operate in high intensity radiation fields. The properties of polymeric materials can be greatly affected.

Reliability: It is often mandatory that valves operate perfectly for thousands of cycles at unattended industrial plants.

The problems enumerated encompass nearly every facet of valve design. The majority of problems, however, occur in three areas: extremely low and high temperatures, and reliability. The primary symptom of unreliability is leakage, which generally results from contamination in the system.

New materials, fresh design thinking, uncommon ingenuity are reflected in today's valves, actuators and positioners. The valve has made more progress in the last two decades than in any similar span of its 2000 years of development. The most apparent indication of genuine design advance is a graceful external appearance that "looks right", the typical modern valves make this point. Let's briefly review the conditions under which the valve lives.

Physical loads of valve come from line fluid and from adjoining pipes. Fluid pressure

range from high vacuum, causing light external load, to steady internal pressure of $3.45*10^7$ N/sq m or more. Fluid-transmitted shock and hammer can produce pressure spikes far in excess of nominal pressure maximum. Loads from adjoining piping can impose another serious threat to valve tightness and ease of operation.

Other enemies of valves include heat, cold, cavitation, corrosion, and erosion. Designer combat high temperature with appropriate alloys. Cavitation and erosion are two old foes that are being fought with increasing success today. Special disc materials and seat surface hardness can exceed 700 Brinell in today's valves. Stelliting is becoming more common for primary seating surfaces. Flow passage design continues to improve. It reduces erosion that has been a trouble source in many services, from feed water to hydraulic servo. Maintainability is better and still improving. Most of the new valve and actuator designs facilitate rapid inspection, repacking, regrinding and replacement. Control valves pioneered some of this improvement with quick-change trim.

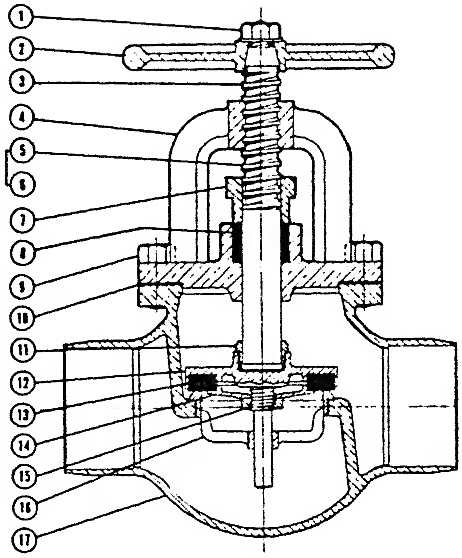
1. 3 GLOBE Vs ROTARY VALVES

There has been a virtual revolution in the design and application of control valves. The once popular single seated and double seated valves have been displaced by top entry cage and flangeless rotary valves as the top choice in the field .The double seated valve, which only five years ago accounted for 50% share of the market, now account for less then 3% .

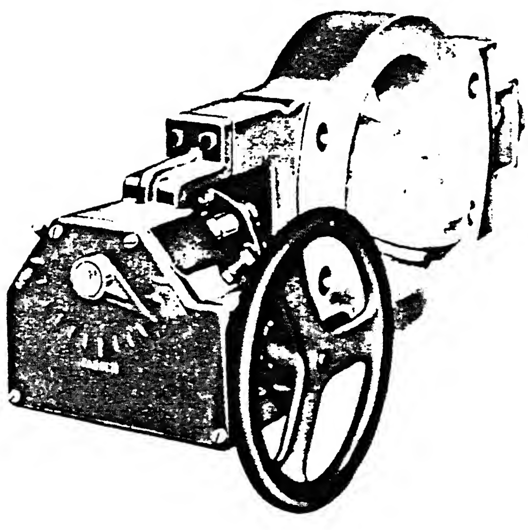
The modern cage valve has certain advantage such ease of maintenance, adaptability to low-noise trims, and better guiding. But the cage valve has the disadvantage of cost in the larger valve sizes. Figure 1.1 shows difference between globe and rotary valve, and figure 1.2 illustrates the cost advantage that the flangeless rotary valve (including butterfly valves) have over globe valves. Especially, in size above 100 mm rotary valves have the ability to offer tight shutoff with much less actuator power. They are small in size and lighter weight especially in size above 75 mm. There is clearly a size and weight advantage of a rotary valve over a globe valve for the same size valve.

Now, newer designs offer low-torque valves encapsulated actuators, backslash-free shaft connections, and eccentrically off-set disc with teflon seats for tight shut-off. The result is a dramatic upward shift in the market share enjoyed by a butterfly valves.

PART
1. Handwheel Nut
2. Handwheel
3. Stem
4. Bonnet
5. Gland Nuts
6. Gland Studs
7. Gland
8. Packing
9. Hex Head Capscrew
10. Body Gasket
11. Disc Holder Nut
12. Disc Holder
13. Disc
14. Disc Plate
15. Seat Disc Nut
16. Seat Ring
17. Body



Globe-type valve with flat-face bolted bonnet.



Courtesy of Continental Div. Fisher Controls Co.

Handwheel-actuated butterfly valve.

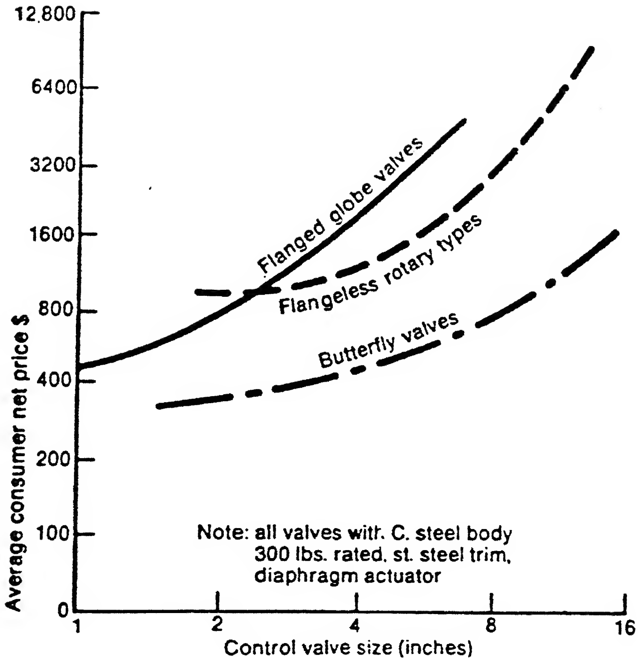


FIGURE 1.2 : Flangeless rotary valve,particularly Butterfly valves, Have a greater cost advantage over Globe Valves Reference[3].

1. 4 GROWTH OF HIGH PERFORMANCE BUTTERFLY VALVES

There have been several important changes in the application of valves by most industrial users. These developments includes the following.

1. Process systems have increased in size, pressure, and temperature.
2. The increased sophistication of production process combined with escalating labour costs have resulted in a growing need for more automated valve systems.
3. Increased legislation has forced users into tighter control of fluids to prevent leaks, spills, and fugitive emissions.
4. Energy costs have contributed to recycling of steam, water,a nd waste heat. This has increased the demand for leak-tightness of valve seats, gaskets, and packing.

The conventional metal-seated valves were found to be deficient for the following reasons:

1. **Cost** As the size of piping system increases, the size and cost of gate, globe, or plug valves tends to increase proportionally faster than line size.
2. **Sealing** A metal-seated valve cannot give completely reliable shutoff. Natural rubber can not be used in because it may deteriorate in time, particularly in contact with petrochemicals.
3. **Automation** Globe and gate valves require many revolutions of the handle or operator to open and close them. This makes them particularly a difficult and expensive to automate.
4. **Throttling** In the partially open position, most gate valves make a poor throttling device because of the chattering created across the valve discs. Globe valve tends to become very expensive as the line size increases, and plug valves have characteristically high torque, which contributes to poor control of the valve.

Over the past several years, their has been a significant change in the industrial valving requirements. A new generation of valves has been introduced based on the development of fluorocarbons such as teflon as a seat material and the acceptance of the butterfly-style body a process valve, a new concept has been developed - The high performance butterfly valve.

Now butterfly valves are being used from low pressure water service to applications in extremely sophisticated valving systems with very high temperature, high vacuum, low temperature applications, very high pressures.

To summarize, it can be said that the high performance butterfly valve market is the fastest changing segment of valve industry, growing at twice the rate of the total valve industry.

It is expected that in 5 years the unit sales will equal those of ball valve market, and that the majority of people now using plug, ball, and gate valves will be intimately familiar with high performance butterfly design.

1. 5 PROBLEM DEFINITION

The details of valve design can be divided into three individual design problems. Each phase or problem defined below can be treated separately .

1. The most important part of the valve is the flow control element. This is that portion of the valve, with its various components, which actually controls the flow of fluids through the valve.
2. Design of mechanism whereby the flow control elements may be adjusted to permit control of flow rate through the valve such as valve actuators, limit switch etc.
3. Isolation of the mechanism governing movement of the flow control elements from the fluid being handled in the valve i.e the sealing methods.

In the present work the first phase has been considered which consist of :

a) Sizing of valve (i.e determining valve diameter for required maximum and minimum flow rate of fluid).

b) Designing major components of valve.

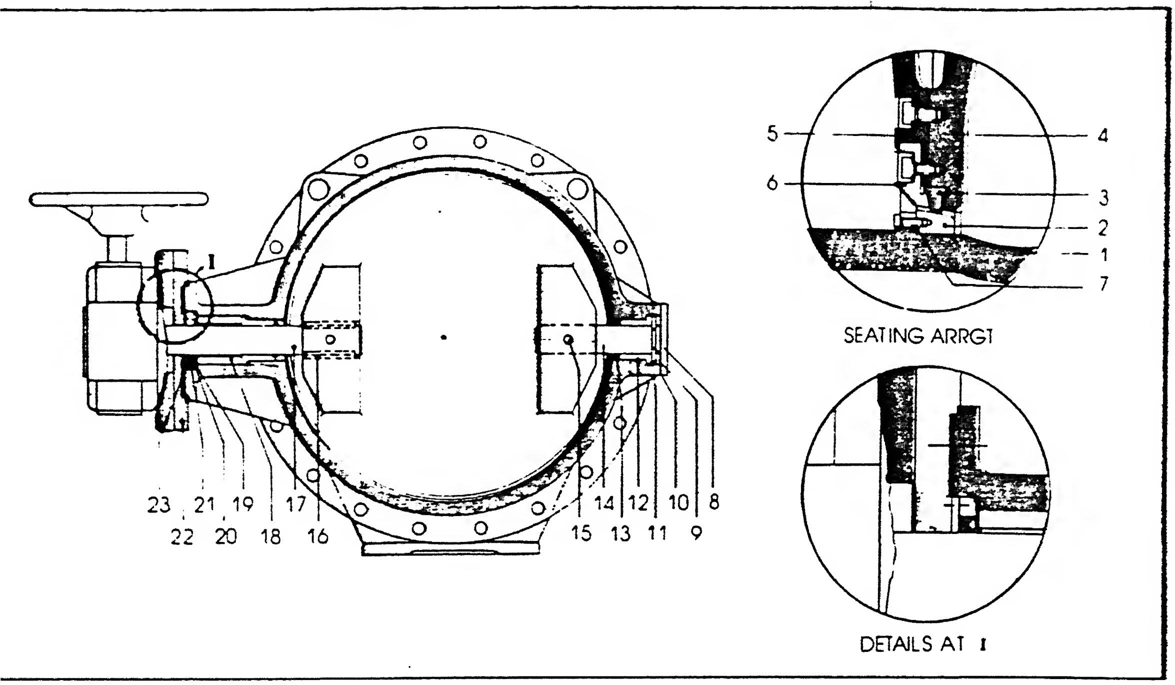
c) Performing stress analysis of the valve components.

d) Preparing final drawing and bills of material.

The design of Butterfly valve requires following specifications from the users.

1. Type of fluids to be handled.
2. Maximum differential pressure across the valve.
3. Maximum and minimum flow rate required.
4. Total head.
5. Type of entrance and exit of the flow.
6. Number and type of bends.
7. Length and type of pipe.
8. Valve flange specifications.
9. Shaft flange specifications.

CROSS SECTIONAL VIEW



PART LIST

Sr No	Description	Material	Sr No	Description	Material
1	Valve Body	C.I IS 210 Gr FG260/300	12	Flanged Bush	Bronze IS:318 Gr LTB-2.
2	Body seat ring	St St B S 970 Gr 316 S 16	13	Packing for Bush	Graphited Asbestos
3	Disc seal	Nitrile Rubber	14	Stub shaft	St. St. BS 970 Gr 431S29
4	Disc	C.I. IS 210 Gr. FG 260/300	15	Dowel Pin	St. St. BS 970 Gr. 316S16
5	Cover for Disc	C.I IS 210 Gr FG 260/300	16	Key for Drive shaft	St St.BS 970 Gr 316S16
6	Clamping ring	M.S.IS 226 Gr Fe410S	17	Drive shaft	St.St.BS 970 Gr 431S29
7	'O' Ring for Body Seat ring	Nitrile Rubber	18	Spacer D.E	M.S.
			19	'U' Cup Seal	Nitrile Rubber
8	Cover for body N.D.E	MS IS 226 Gr. Fe410S	20	'O' ring for 2	Nitrile Rubber
9	Spacer N.D.E	M.S.	21	'U' Seal Retaining Bush	Bronze IS-318, Gr. LTB 2
10	Gasket for Cover & Body	CAF IS 2712 Gr C	22	Adaptor plate	M.S. IS:226 Gr Fe410S
11	Split Thrust Ring	Aust-Iron IS 2749 Gr ASG-Ni-20 Cr-2	23	'U' Seal Retaining Washer	Bronze IS-318 Gr 2

FIGURE 1.3 : Cross-sectional view of Butterfly valve.

All the above conditions are dependent on the requirements of users of the valve. The sizing of valve (i.e determining valve diameter to meet required flow rate) is done by Nomograph technique described later. It involves a number of manual iterations, look up charts etc.

Design of major components of valve and its stress analysis is done by manual calculations. The design methodology of parts is discussed in chapter 3. Following are the parts which are designed step by step. The cross sectional view of the valve is shown in figure 1.3.

DISC : This is the main flow controlling element, which rotates on either vertical or horizontal axis. When the disc lies horizontally, the valve is fully open, and when the disc approaches the vertical position, the valve is shut off. Intermediate position of disc is used for throttling purposes.

WALL BODY: This is the outer body of the valve which houses the disc, shaft, bearing etc. It is generally made of cast iron.

SHAFT: The disc is mounted on the shaft and total torque is transmitted to the gear box through it.

BEARING: The bearing is of journal type with bronze lining. It is designed to withstand full differential pressure on disc.

KEY: The key used is rectangular sunk key for torque transmission between the disc and the shaft.

SPLIT THRUST RING : This is used at the non-driving end of shaft to prevent axial movement.

COVER FOR NON DRIVING END.

LOCKING CAP: It is used to keep the disc in fully open or close position in the event of necessity to remove gear box for repair.

BOLTS FOR LOCKING CAP

VALVE SHAFT FLANGE ANALYSIS

SHAFT FLANGE STRESS ANALYSIS

1. 6 LITERATURE SURVEY

The NOMOGRAPH TECHNIQUE for sizing butterfly valve was first published in [9]. The same has been improved and was made available in [3] from where it has been adopted.The various parts of butterfly valve such as disc, shaft, key, bearing, split ring etc have standard

design procedure and different text books [1,2,3,6,7,12,13,14,15,16] have been referred for design. Every country has its own standardization scheme. Standard data for mechanical components has been referred from ASME (American society of mechanical engineers), B S (British standard) code, I.S.I (Indian Standards Institution) code. General methods for calculating torque values, ADAMS CHARTS (attached as appendix C) for seat torque and shaft diameters for various valve size was made available from [8], a leading manufacture in butterfly valves.

Using many of the above sources, a consolidated approach for the design and analysis of butterfly valves has been developed in this thesis. A computer aided design approach facilitates in automatically generating the basic design of the valve components and its assembly. It will help the designer in iteratively changing the parameters and arriving at a satisfactory design in a shorter time. It is hoped that methodology given here will of help to the manufacturing industries.

1. 7 OBJECTIVE

With the acceptance of butterfly valve in the process control and with the market growing at twice the rate of total valve industry, a need to develop a computer aided design of butterfly valve was felt necessary since the manual designing methods take enormous time (about 500-700 man hours)and any modification or iteration delays the process further.

Keeping in view the above mentioned point, it was felt that an interactive package for the design of all the major components of butterfly valve will significantly enhance the designing capabilities of a designer.

A interactive computer program has also been developed for the Sizing of butterfly valve and design of all the major components of valve using Indian Standards code, B.S specifications and also to carry out stress analysis of components for safety of design, Finally to prepare bills of material and to assemble all the designed component to produce final scale drawing of the valve.

CHAPTER-2

FLOW THROUGH BUTTERFLY VALVES

2.1 INTRODUCTION

The use of a butterfly valve as a final control element is basically associated with a flow control problem, when the object is to maintain

a) Fixed flow rate (flow rate control).

b) Desired line pressure(Line pressure control) .

Figure 2.1 shows details of a typical butterfly valve. The control valve is used as a variable restriction (analogous to variable resistance) to add loss in the system and to establish a energy balance required to produce the desired result. Consider a common example of a valve being used to control flow in connecting pipeline between two ground sources (figure 2.2). The following factors enter into the problem:

1. Total head or potential energy of the system i.e the difference in hydraulic elevation between the water surfaces of reservoirs.

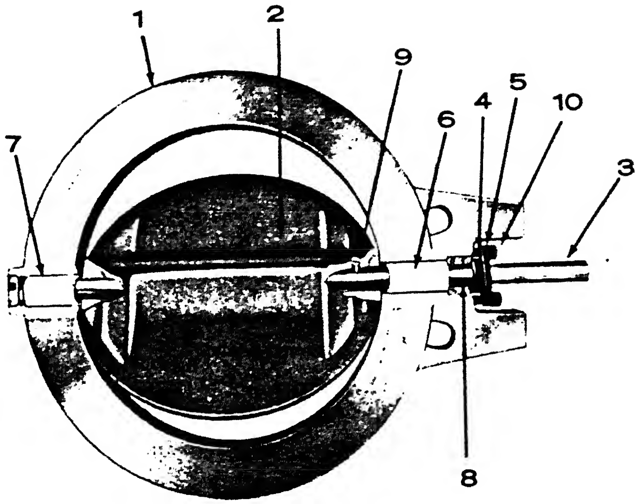
2. Entrance loss as the water leaves reservoirs A and enters the pipeline.

3. Pipe and fitting losses in the line from reservoir A to reservoir B.

4. Exit loss as the water leaves the pipeline and enters reservoir B.

Limitation on total system losses, that is, for a specific flow rate, the total of all system losses (entrance loss at A + pipes & fittings losses + exit loss at B) at that rate must not be greater than the total head or the potential energy available. Any excess of total head (Total head - Head losses in system) available over the system losses for the desired flow rate must be dissipated across the control valve. If the flow rate is changed, the system losses will change and the head to be dissipated across the valve has to change correspondingly.

Entrance and exit losses, pipe and fitting losses, the total head and the types and number of fittings in the line must be known in order to calculate how much energy dissipation (head loss or K factor) is required from the valve.



1—Body
2—Disc
3—Shaft
4—Gland
5—Gland retainer
6—Upper bearing
7—Lower bearing
8—Packing
9—Pin
10—Gland screws

Courtesy of Rockwell Mfg.

Identification of butterfly valve parts.

Courtesy of Continental Div. Fisher Controls Co.

Lever-actuated butterfly valve.

FIGURE 2.1 Details of Butterfly valve.

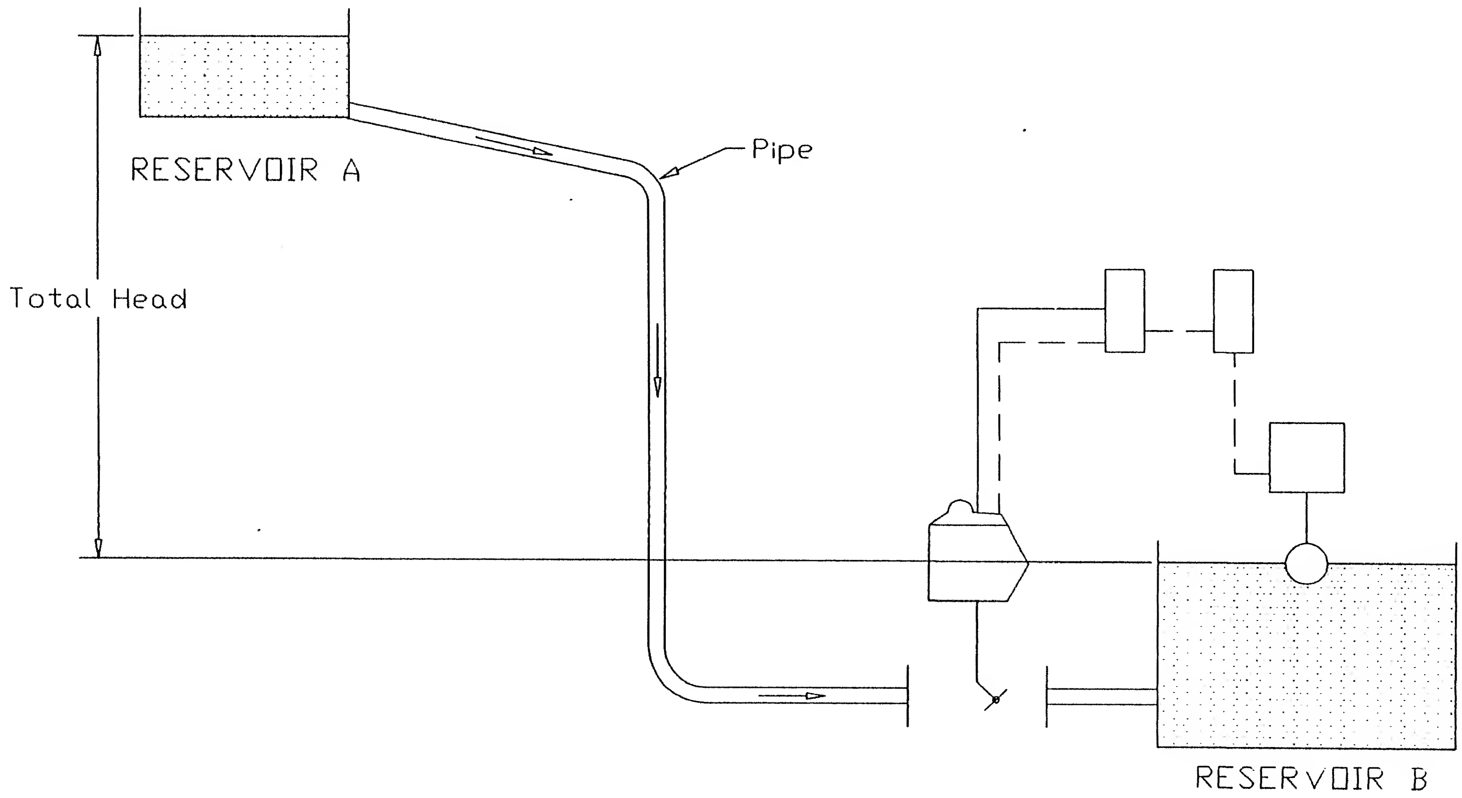


FIGURE 2.2 Instalation employing Butterfly valve for flow control between two ground sources.

The purpose of this chapter is to outline methods for computing pressure drops and hence flow calculation for any type of loss inducer, with particular emphasis on valve sizing. This chapter also present in a convenient format some head loss factor (K) data currently scattered widely through the literature[1,2,3,4,13,14] and to review some background informations on the head loss factor (K) .

2.2 THE HEAD LOSS FACTOR (K-FACTOR)

The K or head loss factor is defined by

$$H_L = K\frac{V^2}{2g} \tag{2.1}$$

The Head loss factor (K) is a measure of the energy (pressure) loss of a fluid passing through a restriction. The velocity to be used in equation (2.1) is taken by convention to be the average velocity through the restriction, which from the continuity equation is a measure of the fluid flow through the restriction .

The Head loss factor (K) is often strongly dependent on Reynolds number and also varies with surface finish in the piping systems and with the geometrical factors, particularly pipe or restriction diameter. It cannot be used in laminar flow regime, since the head loss does not depend on the square of the velocity.

2.3 RELATION TO FRICTION FLOW

The surface finish of flow passage will affect the loss of energy of a flowing fluid. This loss is described in terms of a friction factor, which depends on the surface roughness and the flow area or diameter:

$$f = F(\epsilon, d) \tag{2.2}$$

Where f is the friction factor .

$$K = f\frac{L}{d_L} \tag{2.3}$$

The Head loss factor (K) for a pipe or other fluid conduit depends on this friction factor

$$K = f\left(\frac{L}{d_L}\right)(equivalant) \qquad (2.4)$$

and on the pipe geometry, specifically the length (L) and inside diameter (d_L).

In cases other than straight pipe it is often convenient to express the loss as of an equivalent amount of straight pipe of the same diameter.For example,an elbow may have the same loss factor as 30 diameters length of straight pipe [3].

The friction factor term must be added in to obtain the total Head loss factor (K). Often , however, this term is negligible small compared to other losses and may be ignored.

2.4 RELATION TO d_{EO}, C_V, AND f

In some problem for valve sizing an equivalent sharp edge orifice may be known. In all these relationships d_L is the inlet diameter to the valve or component and b1, b_2 is constant coefficient, d_{EO} is the equivalent orifice diameter. In this case, the total Head loss factor or K-Factor may be found from

$$K = b1\left(\frac{d_L}{d_{EO}}\right)^4 \qquad (2.5)$$

or,conversely, if a total Head loss factor(K) is known, this equation will give the equivalent sharp edge orifice diameter. Similarly , if the valve flow coefficient is known, the total Head loss factor(K) may be found from

$$K = b_2\frac{d_L^4}{C_v^2} \qquad (2.6)$$

The basic problem in valve designing is to properly size the valve to pass a required flow usually with specified pressure drop. Head loss factor (K) analysis provides one way of determining whether or not a specified valve of given size will meet the requirements. From

knowledge of its Head loss factor (K) and pressure drop, an equivalent orifice or valve coefficient can be calculated and used to compute the expected flow.

2.5 PRESSURE DROP

Once the total head loss factor (K) is known from the above three equations or from summing head loss factors or K-factors for individual restrictions, it may be used in Darcy's equation:

$$\Delta P \; \alpha \; \frac{\rho \, q^2(K_{total})}{d_L^4} \qquad (2.7)$$

to find the pressure drop through the components or series of restrictions under consideration.

In many problems of valve design it is desirable to have as low a pressure drop through the valve as possible consistent with the flow requirements.In such cases it is generally desirable to keep the effective flow area between 100 and 120 % of the inlet area [3] because this will result in lowest pressure drop. When a lower pressure drop is required than that obtained, the restrictions with the largest head loss factor may be modified to result in a lower overall K-factor and pressure drop.

2.6 CHANGE IN DIAMETER

In the previous paragraphs it was assumed that the flow diameter never decreases below that of inlet diameter.In those instances the Head loss factor (K) for the various restrictions may be summed to give a total K-factor , which then can be used to find the pressure drop.(fig 2.3 a). In case where a diameter becomes less than the inlet diameter (figure 2.3 b), a new procedure is required. For such problems, the following method is used [3].

The system is partitioned into sections such that for each section the K-factor may be found. Proceeding from the inlet to the outlet, sum all the K-factor for which the diameter is equal or exceeds the inlet diameter and find the pressure drop to this point. For a diameter less less than inlet diameter, the head loss factor (K) is found from that sections to the next one with diameter less than its diameter and pressure drop is computed. At the end sum all the pressure drop to give a total pressure drop for the system."

Thus, equation (2.7) or an equivalent will be used repeatedly, once for each region up to

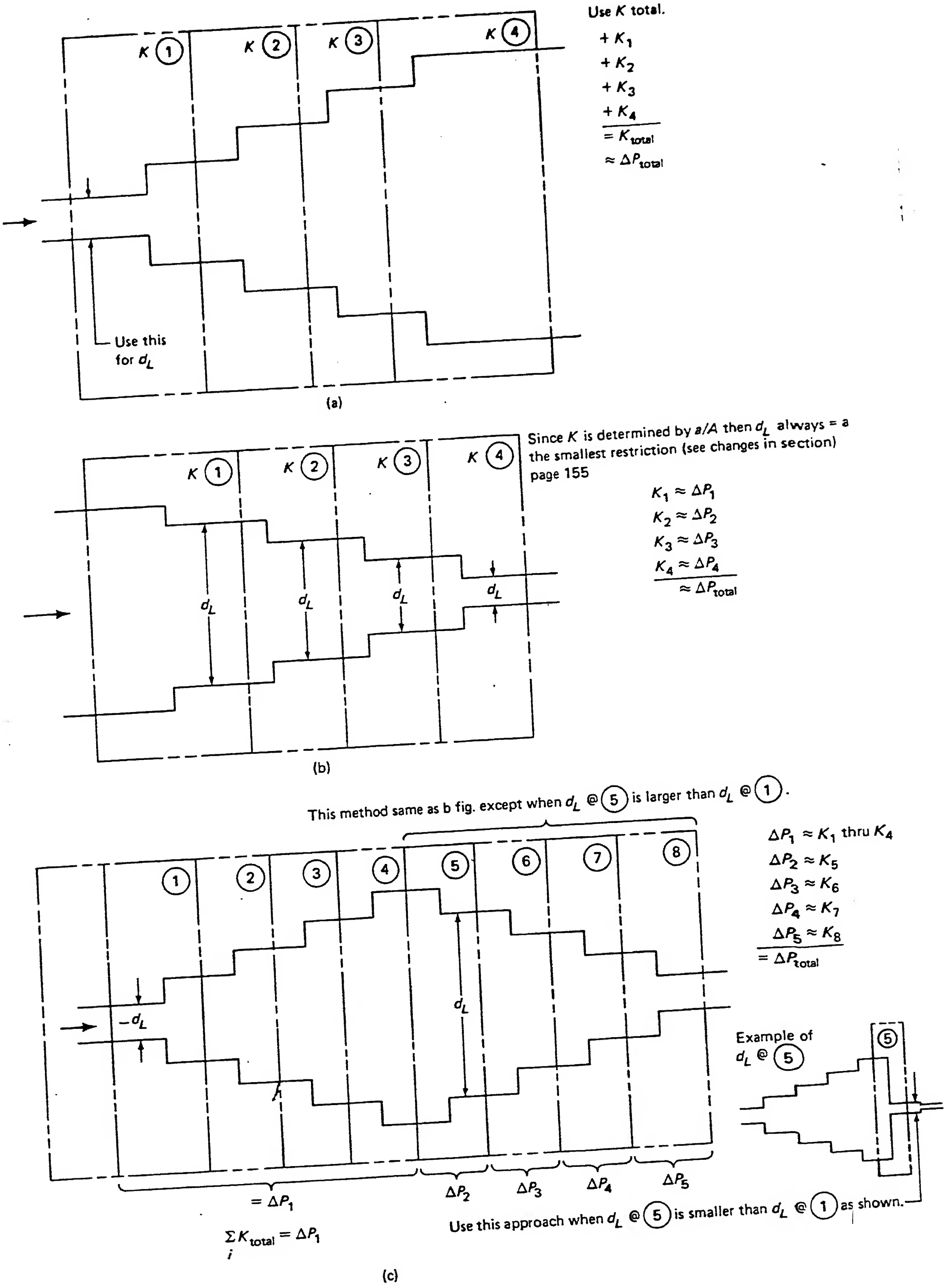


FIGURE 2.3 Calculation of K-factor for change in diameter.

Sudden changes in section

K	0.81	0.64	0.49	0.36	0.25	0.16	0.09	0.04	0.01
a/A	0.1	0.2	0.3	0.4	0.5	06	0.7	0.8	0.9
K	0.4	0.38	0.34	0.30	0.24	018	0.1	0.05	0.015

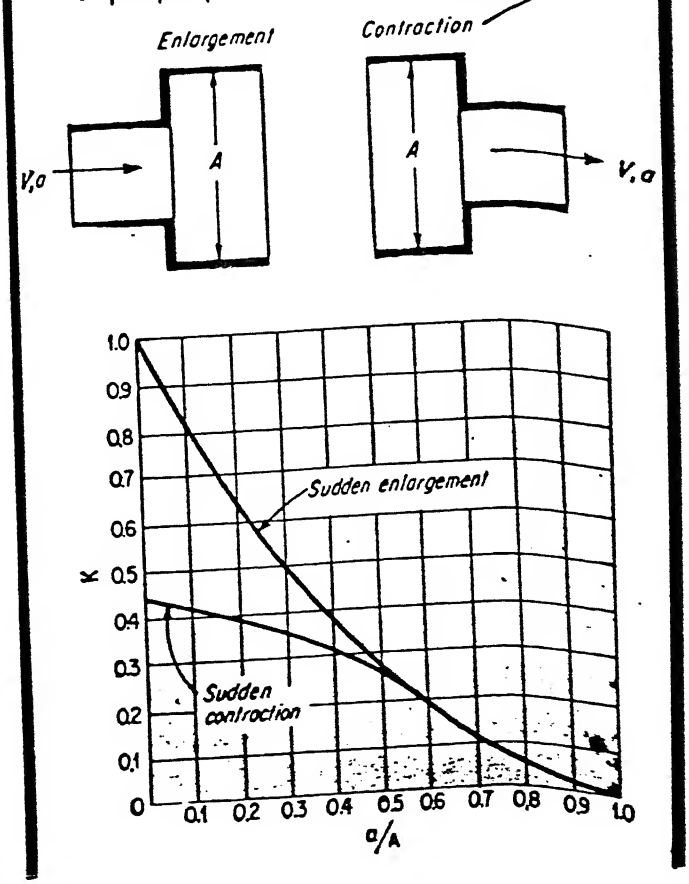


FIGURE 2.4 K-factor for sudden change in section.

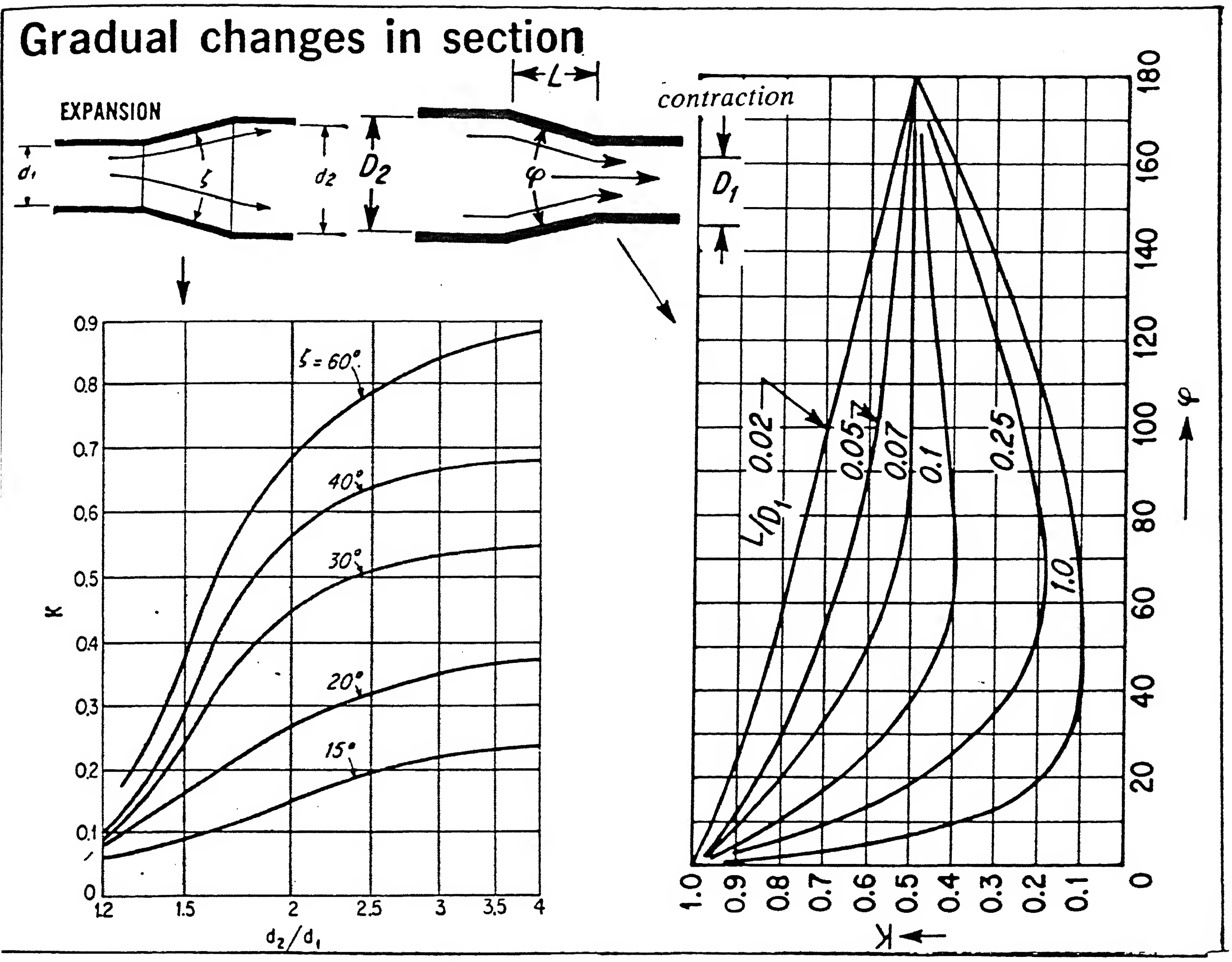
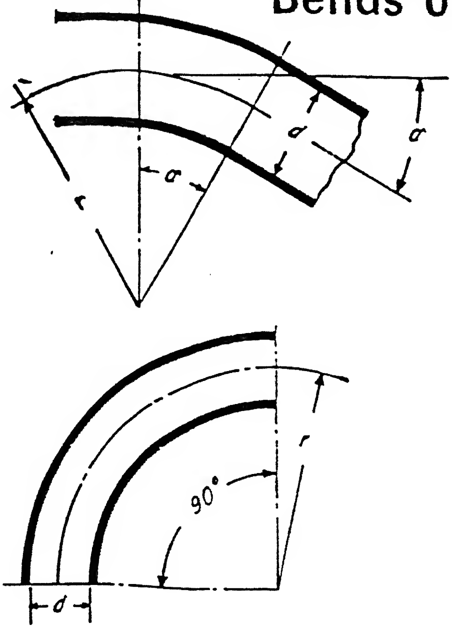


FIGURE 2.5 K- factor for Gradual change in section.

Bends of tubing and hose



Best K = 0.1 to 0.15 for smooth 90°-bend, and r/d at ideal 2 to 4

If bend is angular (or smooth with $r/d < 1$ or > 6) use this table for K: (conservative)

α	10	20	30	40	50	60	70	80	90
K	0.04	0.1	0.17	0.27	0.4	0.55	0.70	0.90	1.12

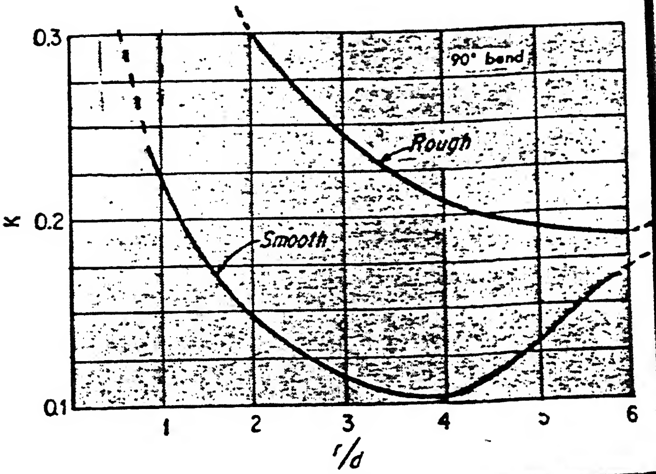


FIGURE 2.6 K-factor for bends of tubing and hose.

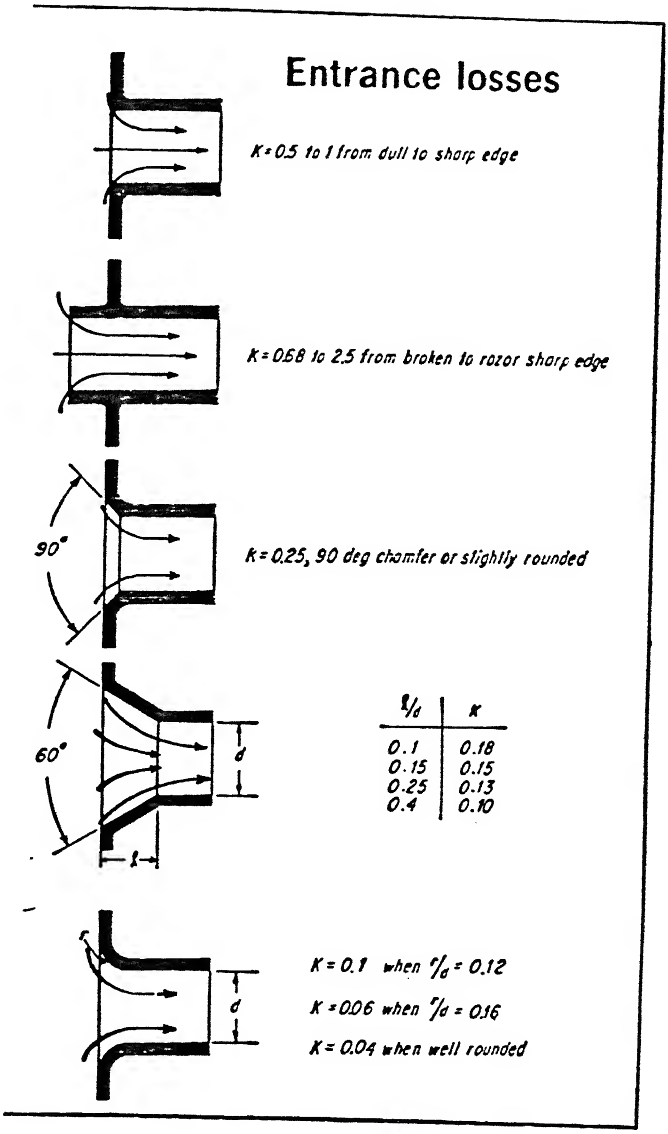


FIGURE 2.7 K-factor for entrance loss.

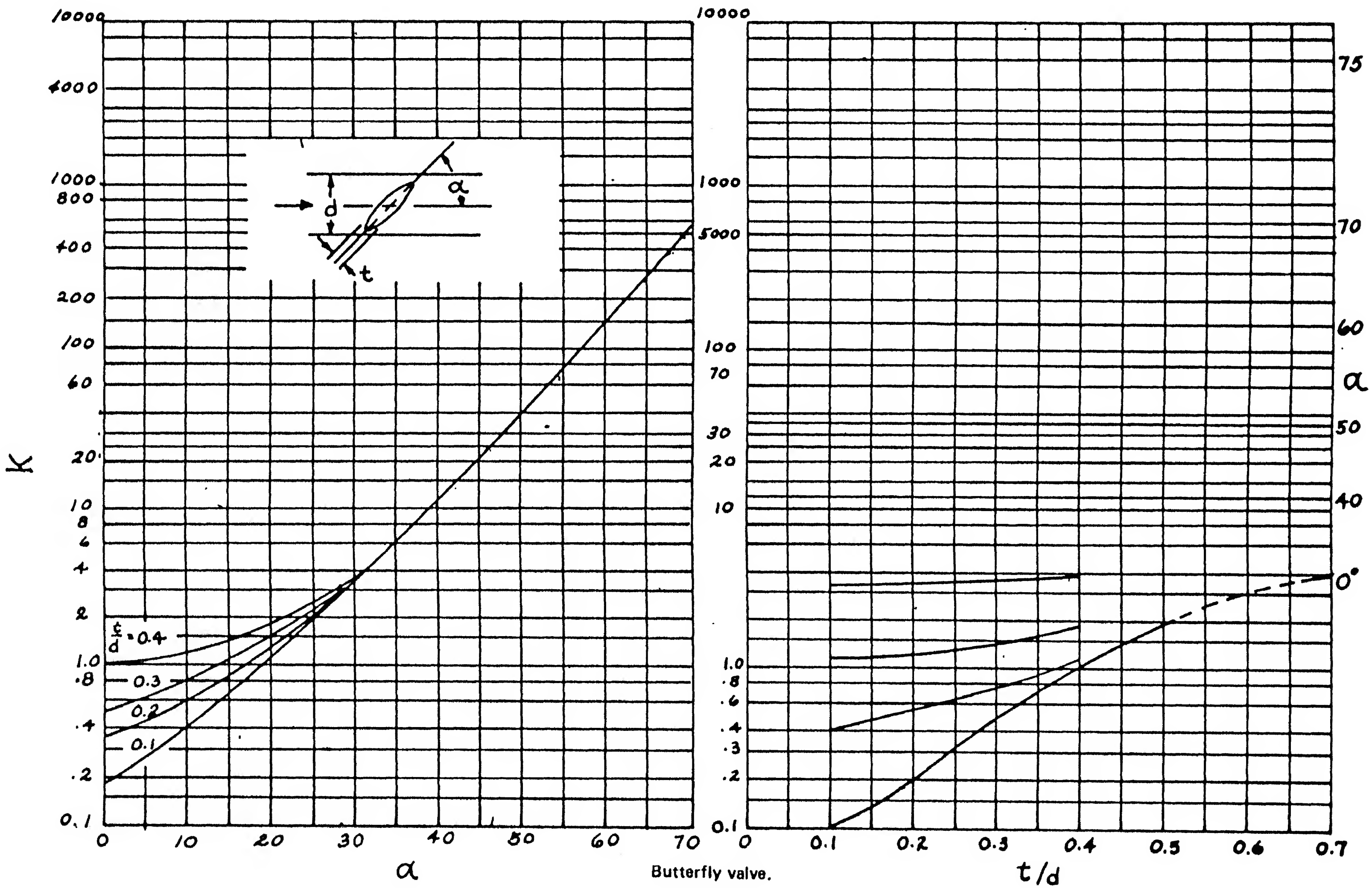


FIGURE 2.8 K-factor for butterfly valve for various disc positions.

a point where the flow diameter decreases to less than the inlet diameter, and these results will be summed up to give the total pressure drop:

$$\Delta P_{total} = \Sigma \Delta P_i \qquad (2.8)$$

where

$$\Delta P_i \approx (\Sigma K_j)_i \qquad (2.9)$$

The following is the K-factor information found widely scattered in the literature [1,2,3,4,13,14] which has been assembled here in a convenient format for use by valve designers and others.

SUDDEN CHANGE IN SECTION: Figure 2.4 shows K-factor for sudden expansion or contractions. For either case the velocity to be used in eq.(2.1) corrosponding to the velocity in smaller diameter section.

GRADUAL CHANGE IN SECTION : Figure 2.5 presents the K-factor for gradual expansion and gradual contraction respectively.

SMOOTH BENDS : Figure 2.6 shows K-factors for smooth 90-degrees bends and correction factor for other angles.

ENTRANCE LOSSES: The K-factors for several entrance configurations are shown in figure 2.7

EXIT LOSS : The K-factor for any exit loss regardless of the exit configuration is 1.0. This follows from the conservation of energy principle.

BUTTERFLY VALVE: Figure 2.8 give K-factor for butterfly valve as a function of amount of opening. Valves may differ from one manufacturer to another.

CHAPTER-3

DESIGN METHODOLOGY FOR BUTTERFLY VALVES

3.1 SIZING OF BUTTERFLY VALVES

Sizing of a butterfly valves basically means determining the diameter of the valve disc to adjust between the required maximum and minimum flow rate. The following information is needed for the sizing of butterfly valves.

a) Maximum flow rate required.

b) Minimum flow rate required.

c) Total head or potential energy of the system.

d) Head loss in the system.

The head loss factor or K-factor (energy loss of fluid passing through restriction) of the system is determined as described in Chapter-2. Then the NOMOGRAPH TECHNIQUES for sizing the butterfly valve for best control operation is used. All applications should be checked for cavitation.

With reference to figure No-3.1, for K=16, it may be noticed that for a disc position of 60 degrees there are relatively small changes in flow for increasing angular travel of disc. A similar situation exists below 15 degrees disc position.

Operation of a valve in the ranges beyond these extreme disc positions tends to produce "mushy"wandering control. Therefore, it is recommended that the butterfly valve be so chosen to operate between angles which are best for the system conditions. This frequently, but not always results in a valve smaller in size. Also each system will have its own head loss factor or K-factor value. Consequently, each system will have its own optimum disc angle.

The rule of thumb that the valve should be sized to have a maximum velocity of approximately 16 ft/sec(4.5 m/sec) is still valid for rough checking. Valve velocity means the velocity through a pipe of the same nominal size of valve.

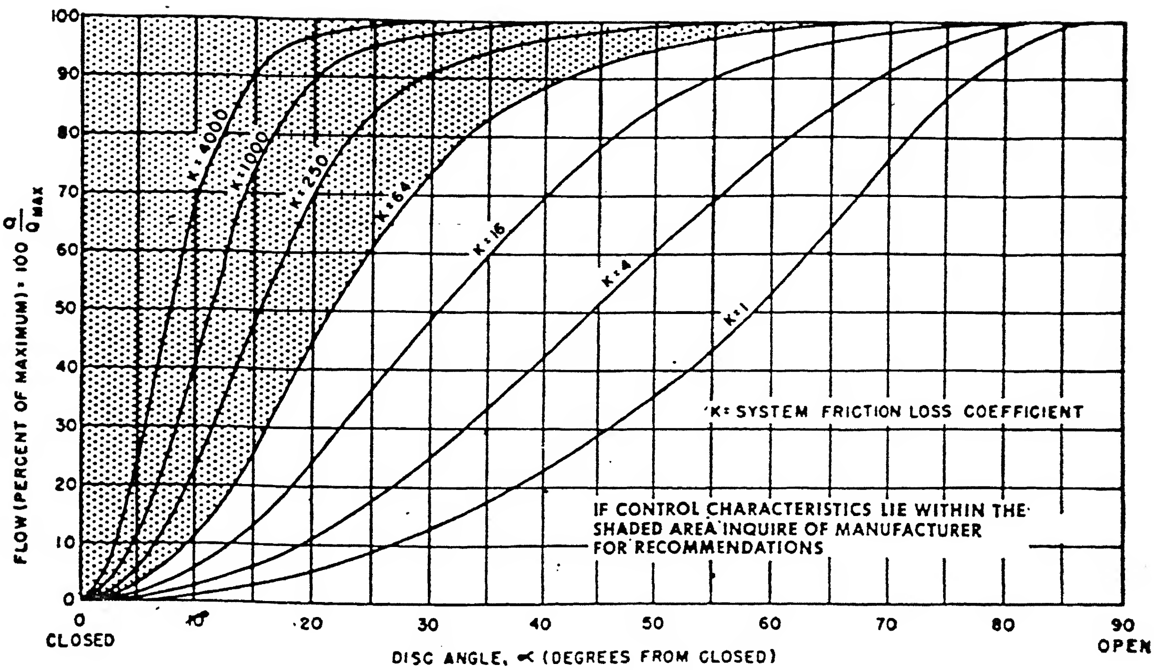


FIGURE 3.1 Relation of flow through butterfly valve Vs disc angle.

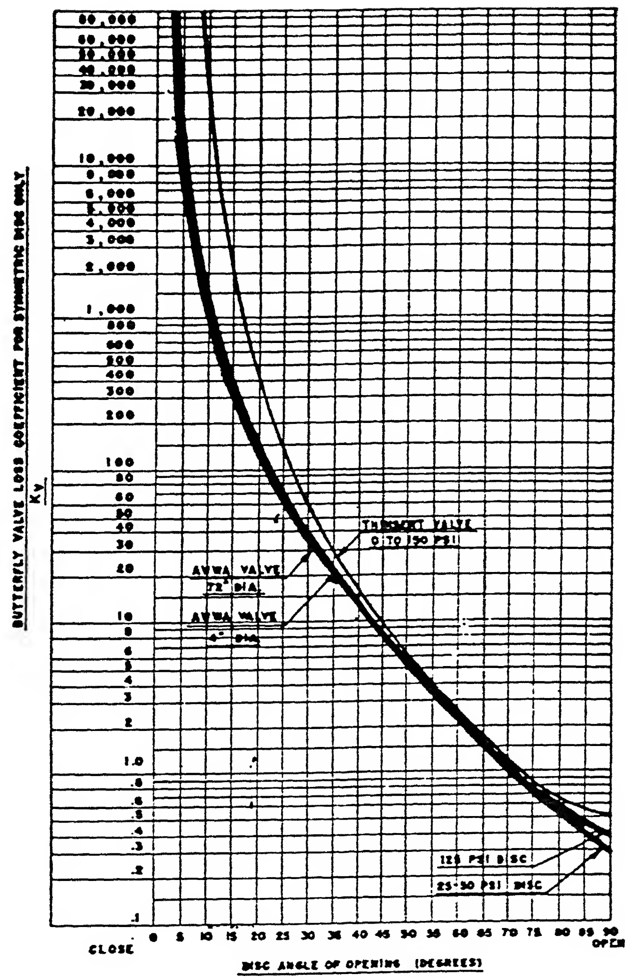


FIGURE 3.2 Relation of butterfly valve loss coefficient to disc angle of opening in degrees.

The nomograph method for sizing of butterfly valve is described below.

3.1.1 NOMOGRAPH TECHNIQUE

The nomograph technique utilizes figures 3.1 & 3.2 . A specific solution of valve size may be worked out using following data:

Step 1.

Diameter of pipeline=1000 mm

Maximum velocity = 2.13 m/sec

minimum velocity = 0.3657 m/sec

Total head (51.8 - 42.67) = 9.13 m

Find System Loss coefficient "Ks"

Entrance Loss(sharp edge)	K=0.50
Exit Loss(sharp edge)	K=1.00
For two 80 mm elbow loss(long sweep)	
(when L/D=20), 0.023*20*2	K=0.92
Riveted steel pipe(f * L/D)	
(when L/D=500), 0.023*500	K=11.50
Total System loss coefficient	Ks=13.92

A curve for Ks=13.92, on figure 3.1 would fall between k=4 and k=16, with optimum control angles from 15 degrees to 65 degrees.

Step 2

Find 1000 mm valve loss(Kv) and opening angles.

Calculate total system Kt, required to dissipate 9.13 m of differential head.

<u>a) FOR MAXIMUM FLOW</u>

$Kt=\Delta H*2*g / V^2 max = 9.13*2*9.82 / (2.13)^2$ =39.48

$Kv=Kt - Ks = 39.48 - 13.92$ =25.56

Determine valve open angle from figure 3.2

α max = 33 degrees.

This is an acceptable disc position.

b) FOR MINIMUM FLOW

Kt min = Δ*2*g / V max = 9.13*2*9.81 / $(0.3657)^2$ =1339.42

Required valve loss:

Kv min =Kt -Ks = 1339.42 - 13.92 =1325.5

α min =9 degrees .

This is much less than the recommended 15 degrees.

The 1000 mm valve in this case is not the most satisfactory selection.trying smaller size valve say 600 mm valve size.

The Ks = 13.92 of 1000 mm valve will change as follow:

1000 mm pipe loss(from step 1)	Ks= 13.92
1000 mm to 600 mm gradual contraction (negligible)	
600 mm to 1000 mm gradual enlargement (with total angle of 20 deg)	K=0.25
Total system loss	14.17

Step 3.

Find for 600 mm valve loss Kv and opening angle

a) FOR MAXIMUM FLOW RATE

Required valve loss Kv =

(Kt -Ks) (600/1000) 4 = (39.42 -14.17) (600/1000) = 3.2724

Valve open angle max =52.5 degrees

This is an acceptable position

b) FOR MINIMUM FLOW RATE

Require valve loss Kv min =

(Kt -Ks) (600/1000) = (1339.42 -14.17) (600/1000)4 = 171.75

Valve open angle minimum = 15.5 degrees

This is an acceptable disc position.

The 600 mm valve size would be the most satisfactory selection.

3.2 DESIGN METHODOLOGY FOR THE PARTS OF BUTTERFLY VALVE

In the following sections, design methodology for the parts of butterfly valve is given. The design algorithms along with trial runs is given in chapter 4.

1. a) Valve disc thickness and shape design.

   b) Stress analysis of disc.

2. Wall body thickness design.

3. a) Calculation of shaft diameter and total torque acting on it.

   b) Stress analysis of shaft for combined bending and torsion.

4. Journal bearing design

5. a) Keys for torque transmission.

   b) Stress analysis of key.

6. a) Split thrust ring design.

   b) Stress analysis of split thrust ring.

7. Cover for non driving end.

8. a) Locking cap design.

   b) Bolts for locking cap.

9. Valve flange stress analysis.

10. Shaft flange stress analysis.

3.2.1 CALCULATION OF DISC THICKNESS/SHAPE/STRESS ANALYSIS

The disc is assumed to be a circular plate, Uniformly distributed load, at fullｙy closed position ,and supported at two diametral end by shafts.

The following inputs are required for design of disc.

a) Material for disc.

b) Design factor of safety.

The disc thickness and deflection is calculated from standard formulas of theory of plates.

With reference to figure No-3.3

$$W = P * 0.5 * \pi * D^2. \qquad (3.1)$$

$$Thickness = \sqrt{\frac{k1 * w}{s}} * 10.$$

Maximum deflection at mid point of circular edge.

$$y = \frac{k2 * w * (0.5 * D)^2}{E * t^3}$$

Once the thickness of disc is calculated from above formula (3.1) the shape determination and stress analysis of disc is done as follows

Reference is made to figure 3.3 for stress analysis of disc of lens shape .

$$R = 0.9 * D \qquad (3.2)$$

$$E = 0.15 * R$$

$$C = 0.75 * D$$

$$S_s = \frac{H}{1.5}$$

continued :-

$$\alpha = 2.\arccos\left(\frac{F+\frac{H}{2}}{R}\right)$$

$$\beta = 2.\arccos\left(\frac{F+\frac{H}{2}}{R+Ss}\right)$$

$$A = 2.R.\sin\left(\frac{\alpha}{2}\right)$$

$$K = 2.(R+Ss)\sin\left(\frac{\beta}{2}\right)$$

$$Ix = \frac{(D-C).H^3}{12} + 2.\left[\frac{(R+Ss)^4}{8}\left(\pi.\frac{\beta}{180} - 0.5*\sin(2.\beta)\right)\right]$$

$$-2.\frac{K^6}{12^2\left[\frac{(R+Ss)^2}{2}\left(\pi.\frac{\beta}{180} - \sin\beta\right)\right]}$$

$$+2.\left[\frac{(R+Ss)^2}{2}\left(\pi.\frac{\beta}{180} - \sin\beta\right)\right].\left(\frac{K^3}{12.\left[\frac{R+Ss)^2}{2}.\left(\pi.\frac{\beta}{180} - \sin\beta\right)\right]} - F\right)^2$$

$$-2.\left[\frac{R^4}{8}\left(\pi.\frac{\alpha}{180} - 0.5*\sin(2.\alpha)\right)\right] + 2.\frac{A^6}{12^2\left[\frac{R^2}{2}\left(\pi.\frac{\alpha}{180} - \sin\alpha\right)\right]}$$

$$-2\left[\frac{R^2}{2}\left(\pi.\frac{\alpha}{180} - \sin\alpha\right)\right].\left(\frac{A^3}{12.\left[\frac{R^2}{2}\left(\pi.\frac{\alpha}{180} - \sin\alpha\right)\right]} - F\right)^2$$

$$Wx = \frac{Ix}{E}.$$

$$M_B = \frac{1}{2}.\frac{(D^2.\pi)}{4}.P.\frac{2.D}{3.\pi} = \frac{D^3.P}{12}.$$

$$\sigma_{vorth} = \frac{M_B}{Wx}.$$

$$\sigma_{zul} = \frac{K}{S}.$$

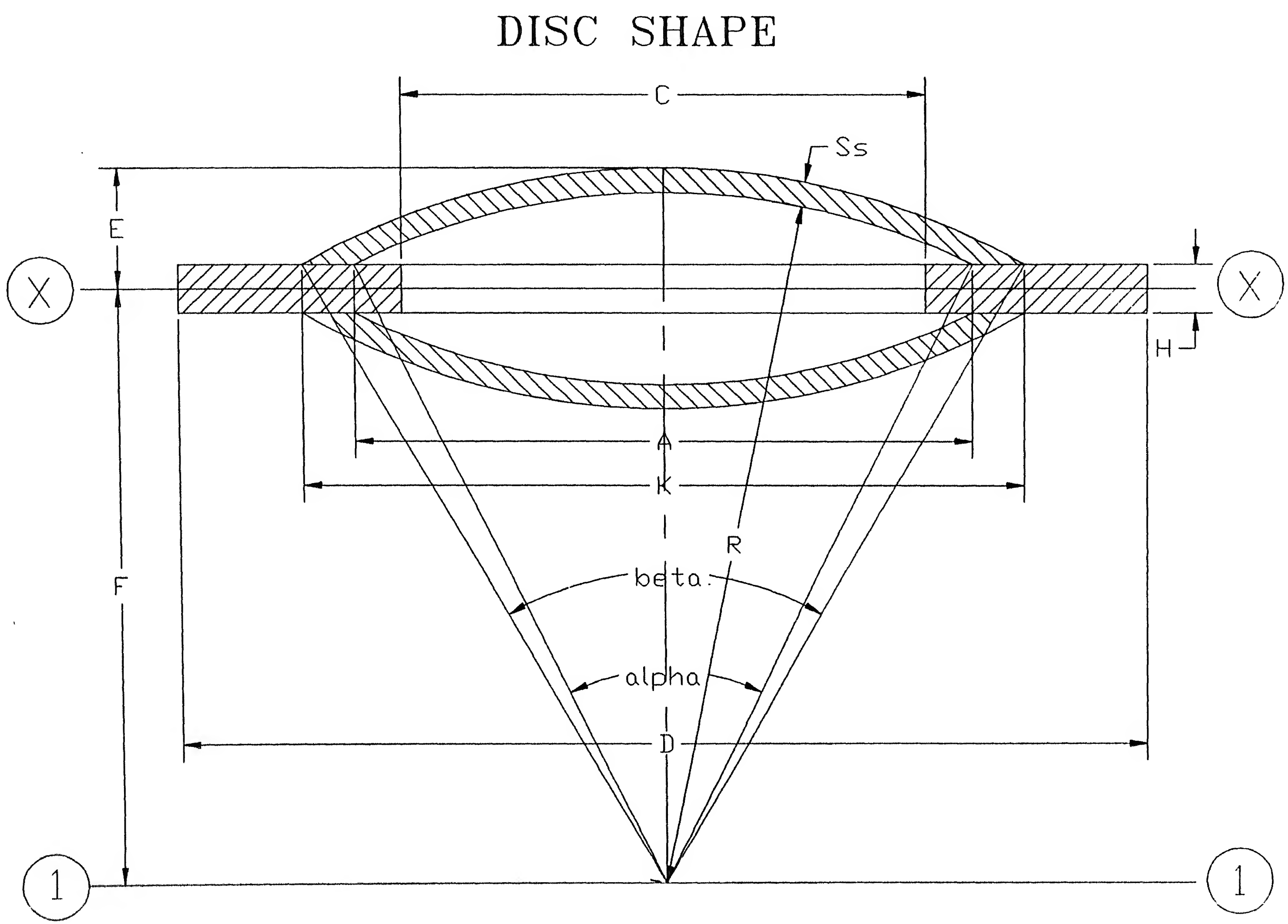


FIGURE 3.3 Lens shape of butterfly valve disc.

DESIGN DATA:-

P(bar)________________Design pressure.

H(mm)________________Thickness of disc.

R(mm)________________Radius of curvature.

y(mm)________________Deflection of disc.

D(mm)________________Diameter of disc.

σ_{zul}(MPa)________________Allowable design stress.

σ_{vorth}(MPa)______________Stress developed in disc.

3.2.2 BODY WALL THICKNESS

Following parameters are required for the design of body wall .

a) Material for wall.

b) Design factor of safety.

With reference to figure-3.4

$$S_r = \frac{D_a.P1}{20.\frac{K}{S}.V+P1} + C1 + C2. \qquad (3.3)$$

$$\frac{D_a}{D_i} \leq 1.2$$

DESIGN DATA

Sr (mm)______________Calculated wall thickness.

Da (mm) ______________Body outer diameter.

Di (mm)______________Body inner diameter.

P1 (bar)______________Design pressure.

K (N/sq mm)__________Design stress value.

S____________________Design factor of safety

V____________________Efficiency factor.

C1(mm)______________Tolerance allowance.

C2 (mm) ______________Corrosion allowance.

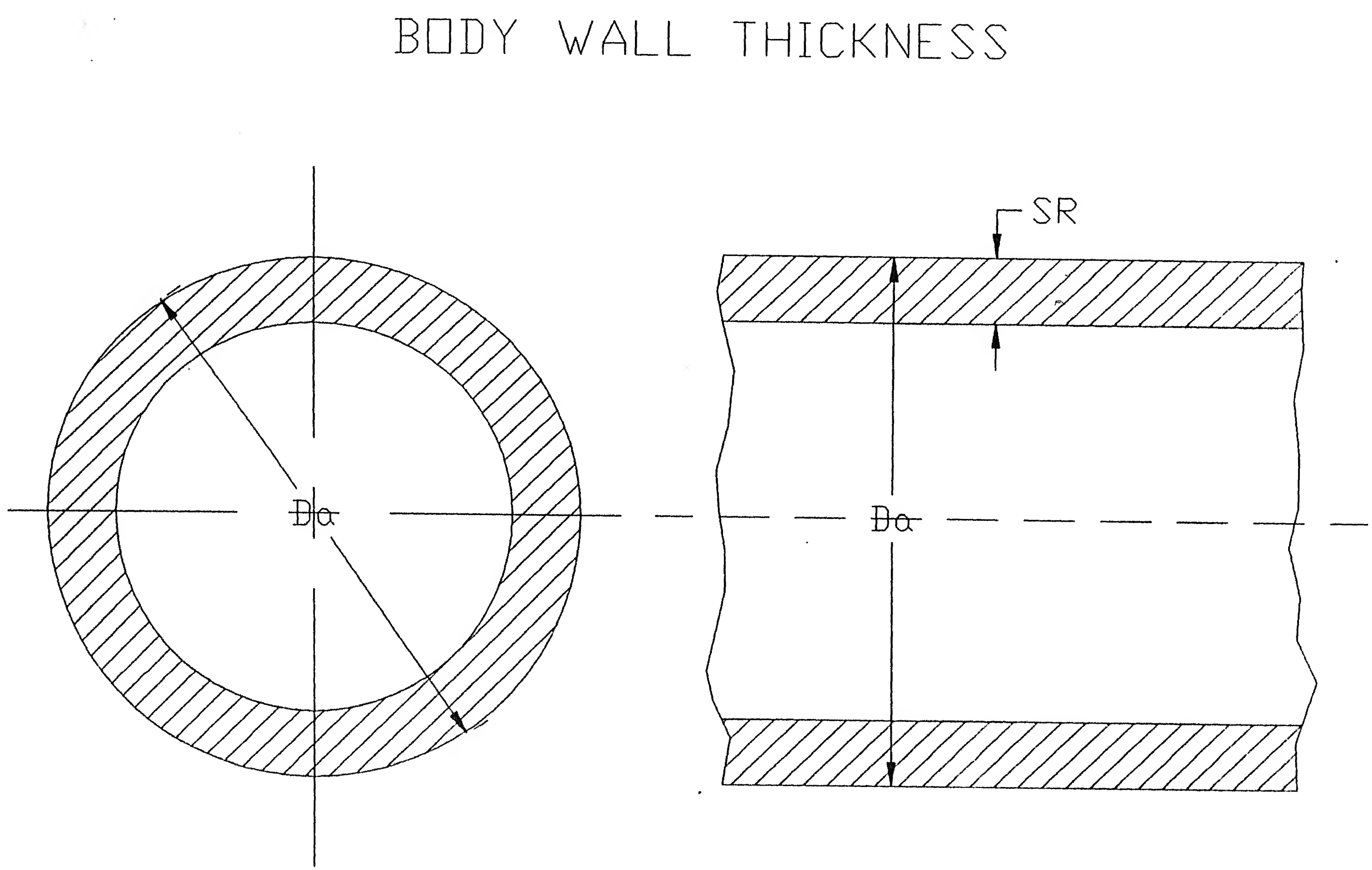


FIGURE 3.4 Sectional view of body wall thickness.

3.2.3 CALCULATION OF SHAFT DIAMETER/OFFSET OF SHAFT/SEAT TORQUE/BEARING TORQUE/OFFSET TORQUE/TOTAL TORQUE

The following inputs are required

a) Material for shaft.

b) Design factor of safety.

1. The shaft diameter (mm) is obtained from ADAMS Chart attached as appendix C .

2. Seat torques(Tseat) as per valve size is obtained from K138 chart of ADAMS attached as appendix C.

3. Offset of shaft (mm) as per valve size is obtained from K138 chart of ADAMS attached as appendix C.

The torques are calculated as follows:-

bearing torque (Tb) =Kr *0.785 * D2 *d/2 * Δ P * μ * 100. (3.4)

Offset torque(Toff) = 0.785 *D2 * offset *ΔP * 100000.

Total torque(NM) =Tb + Toff +T seat.

DESIGN DATA

Kr	Bearing torque coefficient
μ	Coefficient of friction
D	Valve diameter(M)
d	Valve shaft diameter(MM).
ΔP	Working pressure.
Offset (M)	Distance between shaft and body center line.

3.2.4 STRESS CALCULATION OF VALVE SHAFT DUE TO COMBINED BENDING AND TORSION.

The shaft is subjected to combined bending and torsion affect, and the resultant stress is calculated from the following formulae.

With reference to figure No-3.5

$$M_b = \frac{DN^2.\pi}{4} * 0.5 . p2. a. 10^{-4} \qquad (3.5)$$

$$M_v = \sqrt{Mb^2 + 0.75*(\alpha 0.Mt)^2}$$

$$\alpha 0 = 0.7 \, (\textit{Torsion dead} \, V \, \textit{growing, bending changing})$$

$$W = \pi . \frac{d^3}{32}$$

$$\sigma vorth = \frac{M_v}{W} .10^3$$

$$\sigma zul = \frac{K}{S}$$

DESIGN DATA:-

DN (mm)____________________Nominal diameter of valve.

P2(bar)____________________Design pressure.

a(mm)____________________Moment arm.

Mt(Nm)____________________Input torque.

D (mm)____________________Shaft outer diameter.

K (N/sq mm)____________________Design stress value.

S(_)____________________Design factor.

M_b(N-m)____________________Bending moment

σ_{zul} (MPa)____________________Allowable design stress.

σ_{vorth}(MPa)____________________Calculated stress in shaft.

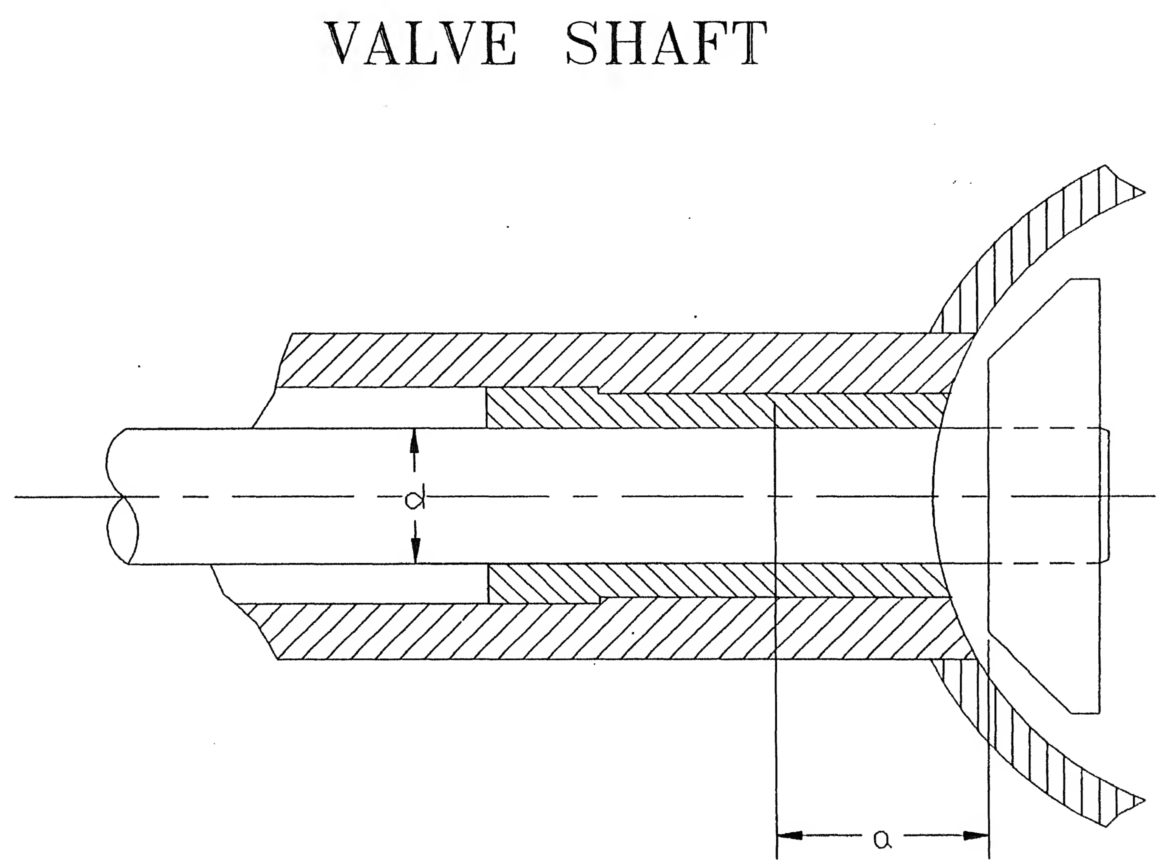


FIGURE 3.5 Sectional view of butterfly valve shaft and bearing.

3.2.5 DESIGN OF SHAFT BEARING AND SHAFT SEALS

The shaft bearing in butterfly valve is of Journal type which are designed to withstand the loads due to full differential pressure on the disc when closed. The bearings are generally made of materials to keep the bearing area absolutely clean and free from any dirt particles the line fluid may be carrying. Provision of 'O' ring or packing ring on the location where the shaft enters body is made. This eliminates the possibility of damage to the smooth bearing area and seizure of the shaft bearing.

For shaft sealing arrangement, valves are provided with unique combination of 'U' cup seal and 'O'ring fitted on retaining bush in such a way that no leakage along the shaft will occur. The shaft seal assembly is easily replaceable at site.

The conventional stuffing box packed with flexible material is generally not used to reduce the operating torque requirement of the valve. It also adds to the difficulty in mounting quarter turn operator. The journal bearing is designed as follow:

With reference to figure 3.5

$F = \pi /4 * D^2 * p$. (3.6)

Bearing length =0.5 *F / bearing stress * d .

Clearance (mm) = 0.02 *d .

Thickness of bush (mm) = 0.08*d +0.3.

Collar diameter =1.2 *d.

DESIGN DATA.

D (mm)________________Diameter of valve.

P (bar)________________Design pressure.

d(mm)________________Diameter of shaft.

3.2.6 DESIGN / STRESS ANALYSIS FOR SHAFT KEYS

Butterfly valve uses SUNK KEY OF RECTANGULAR CROSS SECTION for shaft torque transmissition. This requires the following inputs for design of key :

a) Material for key design.

b Design factor of safety.

c) Numbers of keys .

With reference to figure No-3.6

$$U = d + 13.0 \tag{3.7}$$

$$b = \frac{U}{4}$$

$$L = 1.5 * U.$$

$$h = \frac{U}{6}$$

$$height\ h1 = 0.5\left(\sqrt{d^2 - b^2} - d + 2.t1\right)$$

$$height\ h2 = h - h1.$$

$$surface\ stress \in shaft$$

$$\sigma 1 = \frac{2M_T}{(d - h1).(c - b).h1.n}$$

$$surface\ stresses \in Hub$$

$$\sigma 2 = \frac{2.M_T}{(d + h2).(c - b).h2.n}$$

DESIGN DATA:-

Mt (Nm)________________________ Torque.

d(mm)__________________________Shaft outer diameter.

t1(mm)_________________________ Depth Of shaft key.

H (mm)_________________________Height of key.

L(mm)__________________________Length of key.

b(mm)__________________________Width Of key.

n(_)__________________________ Number of keys.

KEY WITH SHAFT

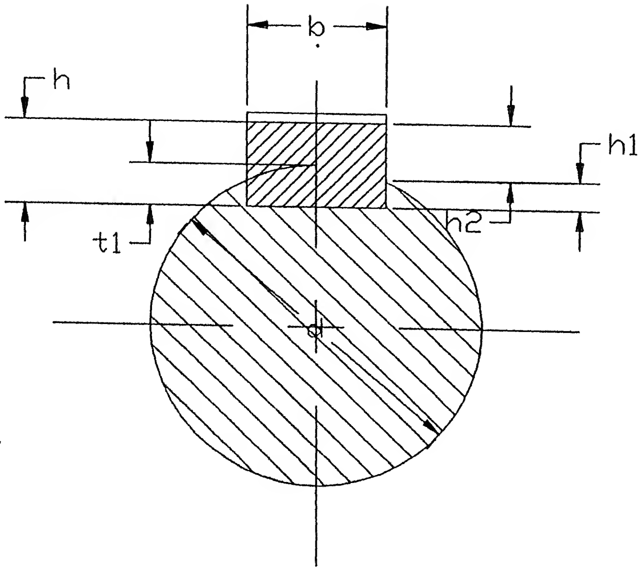


FIGURE 3.6 Sectional view of shaft with key.

SPLIT THRUST RING

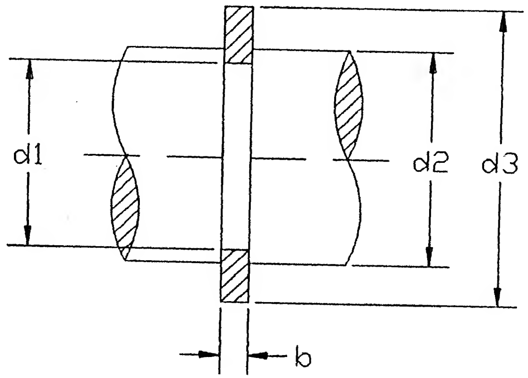


FIGURE 3.7 Sectional view of split thrust ring.

3.2.7 DESIGN AND STRESS ANALYSIS OF SPLIT THRUST RING

Split thrust ring is provided at the non driving end (NDE) of shaft to prevent its axial movements. This requires the following inputs for design of split thrust ring :

a) Material for split thrust ring design.

b) Design factor of safety.

With reference to figure No- 3.7

$$
\begin{aligned}
d1 &= 0.9 * d2. \\
d3 &= 1.2 * d2. \\
\text{Bearing stress} &= \frac{4.\pi.d2^2.p}{4.\pi.(d2^2 - d1^2)} \\
\text{Bending stress} &= \frac{d2^2.(d2 - d1).p}{4.b^2.(d3 - d1)}
\end{aligned} \tag{3.8}
$$

DESIGN DATA:-

d1 (mm) __________ Inner diameter of split thrust ring.

d2 (mm)__________ Shaft diameter.

d3(mm)___________Outer diameter of split thrust ring.

p (Bar)___________Design Pressure.

b (mm)__________ Width of split ring.

3.2.8 DESIGN OF LOCKING CAP / BOLTS FOR LOCKING CAP

The purpose of locking cap arrangement is to keep valve disc locked either in fully closed position or in fully open position in the event of necessity to remove gear box for maintenance while keeping the valve in the pipe line. Locking cap is fitted on valve body for non driving end (NDE) shaft. The locking cap has a suitable bore to receive NDE shaft. Two keys are provided for engaging locking cap with NDE shaft which are normally not engaged during operation of the valve. The disc can be locked in close and open position simply by inserting keys between locking cap and NDE

shaft. Grub screws are also provided to screw the keys in position.

The locking cap is designed to carry full torque. The torque from the shaft goes to the locking cap through keys. This torque is supplied in the form of frictional grip between the locking cap flange and valve flange.The connecting bolts is suitably designed.

The following inputs are required for design:

a) Material for cap and bolts.

b) Design factor of safety for cap and bolts .

c) Total numbers of bolts required.

Torque is transmitted in two ways.

$$r1 = 1.5 * r. \qquad (3.9a)$$
$$r2 = 3.0 * r.$$
$$p.c.d = 2.0 * \left(\frac{r1 + r2}{2}\right)$$
$$shear\,force(F1) = \frac{T * 1000.0}{r1}.$$
$$Thickness(t) = \frac{F}{2.\pi.r1.Design\,stress}$$

*) Frictional torque between two flanges.

**) Shear in bolts.

As a safer assumption it could be assumed that all the torque is transmitted as explained in condition *)

$$Frictional\,torque(T) = \frac{2}{3}.\pi.(r2^3 - r1^3).p. \qquad (3.9b)$$
$$Area(A) = \pi.(r2^2 - r1^2)$$
$$Load(F2) = p.A.N.$$
$$Bolt\,area(A_c) = \frac{F}{N * Design\,stress}$$
$$Core\,diameter(d_o) = \frac{\sqrt{4.0. * A_c}}{\pi}$$

DESIGN DATA

r1(mm)________________Inner radius of locking cap.

r2 (mm)________________Outer radius of locking cap.

T (N-m)________________ Total torque.

F1(N) ________________ Shear force.

F2(N)________________Total load.

t (mm)________________ Thickness of cap.

N(_)________________ Number of bolts.

Ac(sq-mm)______________ Core area of bolt.

dc (mm) _______________ Core diameter of bolt.

3.2.9 STRESS ANALYSIS FOR VALVE FLANGE

Flanges of standard dimensions as per Indian Standards I.S:6392-1971[11] are available in the market. Hence stress analysis of flange of standard dimensions is only required. The following data are needed for stress analysis:-

a) Material for flange.

b) Design factor of safety.

c) Standard dimensions of valve flange as per I.S.code IS:6392-1971

The stress analysis is done as follows:-

Reference is made to figure 3.8

continued on next page:-

$$P_{RP} = p1 . \frac{\pi}{4} . d^2 . 10^{-1} \qquad (3.10)$$

$$P_R = P_{RP} + P_{RZ}$$

$$P_F = p1 \frac{\pi}{4} . \left(d_D^2 - d^2\right) . 10^{-1}$$

$$P_{DB} = p1 . \pi . d_D . k1 . s_D . 10^{-1}$$

$$a_R = 0.5 . (d_t - d - S_R)$$

$$a_F = 0.25 . (2 . d_t - d - d_D)$$

$$a_D = 0.5 . (d_t - d_D)$$

$$M_1 = P_R . a_R + P_F . a_F + P_{DB} . a_D$$

$$a_1 = 0.5 . (d_t - d) - s_F$$

$$P_S = P_{SB} = P_R + P_F + P_{DB}$$

$$M_2 = P_S . a_1 .$$

$$d_L' = d_l \left(1 - \frac{d}{1000}\right) \qquad \textit{for } d < 500 \textit{ mm}$$

$$d_L' = = 0.5 . d_L \qquad \textit{for } d \geq 500 \textit{ mm}$$

$$S_L = \frac{P_R}{\pi . (d + S_R) . K}$$

$$W_{A-A} = \frac{\pi}{4} . \left[(d_a - d - 2.d_L') . h_F^2 + (d + s_F) . (s_F^2 - s_L^2)\right]$$

$$e = \frac{(d_a - d - 2.d_L) . \frac{h_F^2}{2} + (h_a - h_F) . (S_F + S_R) . \left(h_F + \frac{h_a - h_F}{3} . \frac{2 . S_R + S_F}{S_R + S_F}\right)}{(d_a - d - 2.d_L') . h_F + (h_a - h_F) . (S_F + S_R)}$$

$$W_{B-B} = \pi . \left[(d_a - d = 2.d_L') . e^2 = \frac{1}{4} (d + S_R)(S_R^2 - S_L^2)\right]$$

$$W_{C-C} = \frac{\pi}{4} . h_F^2 (d_a - 2.d_L')$$

$$\sigma 1 = \frac{M_1}{W_{A-A}}$$

$$\sigma 2 = \frac{M_1}{W_{B-B}}$$

$$\sigma 3 = \frac{M_2}{W_{C-C}}$$

$$\sigma_{zul} = \frac{K}{S . Z}$$

VALVE FLANGE

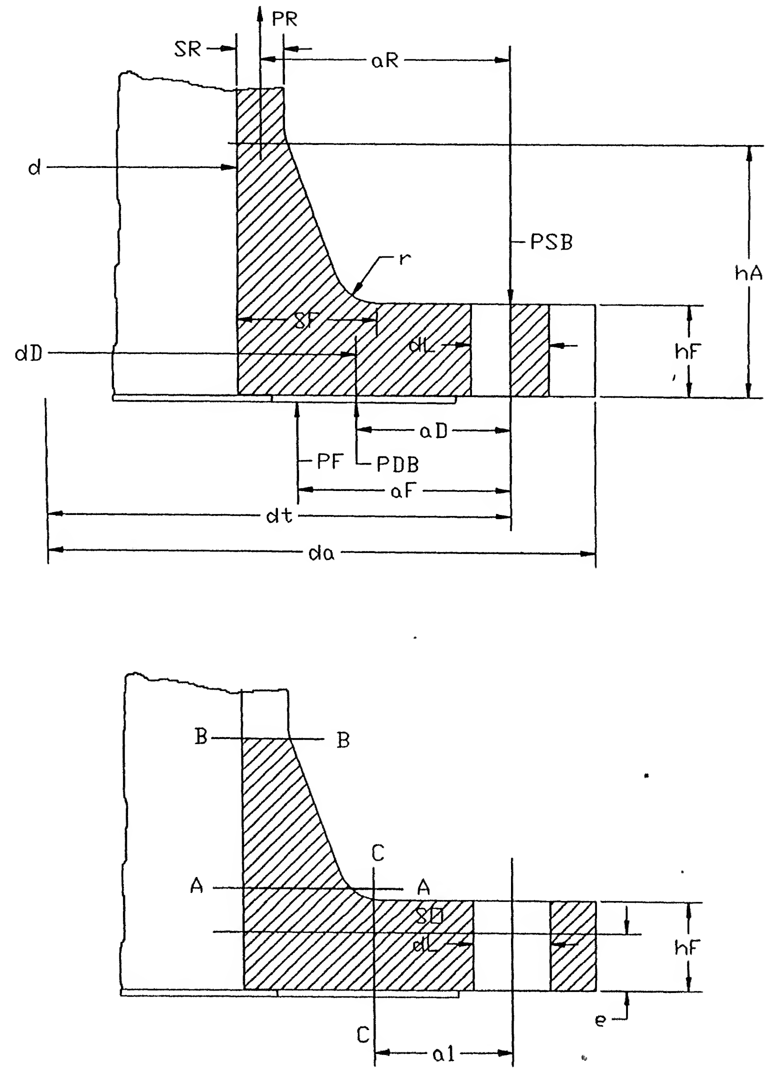


FIGURE 3.8 Sectional view of valve flange.

DESIGN DATA

d(mm)____________________Body/flange inner diameter.

da(mm)____________________Flange outer diameter.

dt(mm)____________________ Bolt circle diameter.

dL(mm)__________________ Diameter of bolt holes.

hF(mm)__________________ Flange thickness.

hA(mm)__________________ Length of flange through hub.

SF(mm)__________________ Thickness of hub at the back of flange.

SR(mm)__________________ Thickness of hub at small end.

K(N/sq mm)______________ Design stress value.

p1(Bar)__________________ Design pressure.

dD(mm)_________________ Diameter atthe location of gasket load reaction.

S(_) __________________ Design factor of safety.

Z(_) ___________________ Factor for correction of allowable stress consdiring deformation properties of material.

P_{RZ}(-)____________________ Additional pipe forces for unspecified conditions.

σ_{zul}(MPa)______________ ___ Allowable design stress.

$\sigma 1$(MPa)___________________Calculated stress in flange through section A-A.

$\sigma 2$(MPa)___________________Calculated stress in flange through section B-B.

$\sigma 3$(MPa)___________________Calculated stress in flange through section C-C.

3.2.10 STRESS CALCULATION FOR SHAFT FLANGE

These are available in standard dimensions as per I.S code IS:6392-1971[ ]. The following input data are required for stress analysis:-

a) Material for flange.

b) Design factor of safety.

c) Dimensions as per I.S.Standards

The stress analysis is done as follows:

With reference to figure 3.9

$$P_R = p1 . \frac{\pi}{4} . (d_i^2 - d_w^2) . 10^{-1} \qquad (3.11)$$

$$a_R = \frac{1}{2}\left[d_t - (d_i + S_R)\right]$$

$$M_1 = P_R . a_R$$

$$d_L' = d_L . \left(1 - \frac{d_i}{1000}\right) \qquad for \quad d < 500\,mm$$

$$S_L = \frac{P_R}{\pi(d_i + S_R).K}$$

$$W_{A-A} = 0.9 . \frac{\pi}{4}\left[(d_a - d_2 - 2d_L')h_F^2 + (d_i + S_R).(S_R^2 - S_L^2)\right]$$

$$\sigma_{vorth} = \frac{M_1}{W_{A-A}}$$

$$\sigma_{zul} = \frac{K}{S.Z}$$

$$W_p = 2.\frac{\pi}{32} . \frac{d_2^4 - di^4}{d_a}$$

$$\tau_{tvorth} = \frac{M_t}{W_p}$$

$$\tau_{tzul} = \frac{\sigma\ 0.2}{2.3}$$

$$\alpha o = \frac{\sigma_{zul}}{1.73 . \tau_{tzul}}$$

$$\sigma_{Vzul} = \sqrt{\sigma_{zul}^2 + 3.(\alpha o . \tau_{tzul})^2}$$

$$\sigma_v = \sqrt{\left(\frac{M1}{W_{A-A}}\right)^2 + 0.75\left(\alpha o . \frac{M_t}{W_p}\right)^2}$$

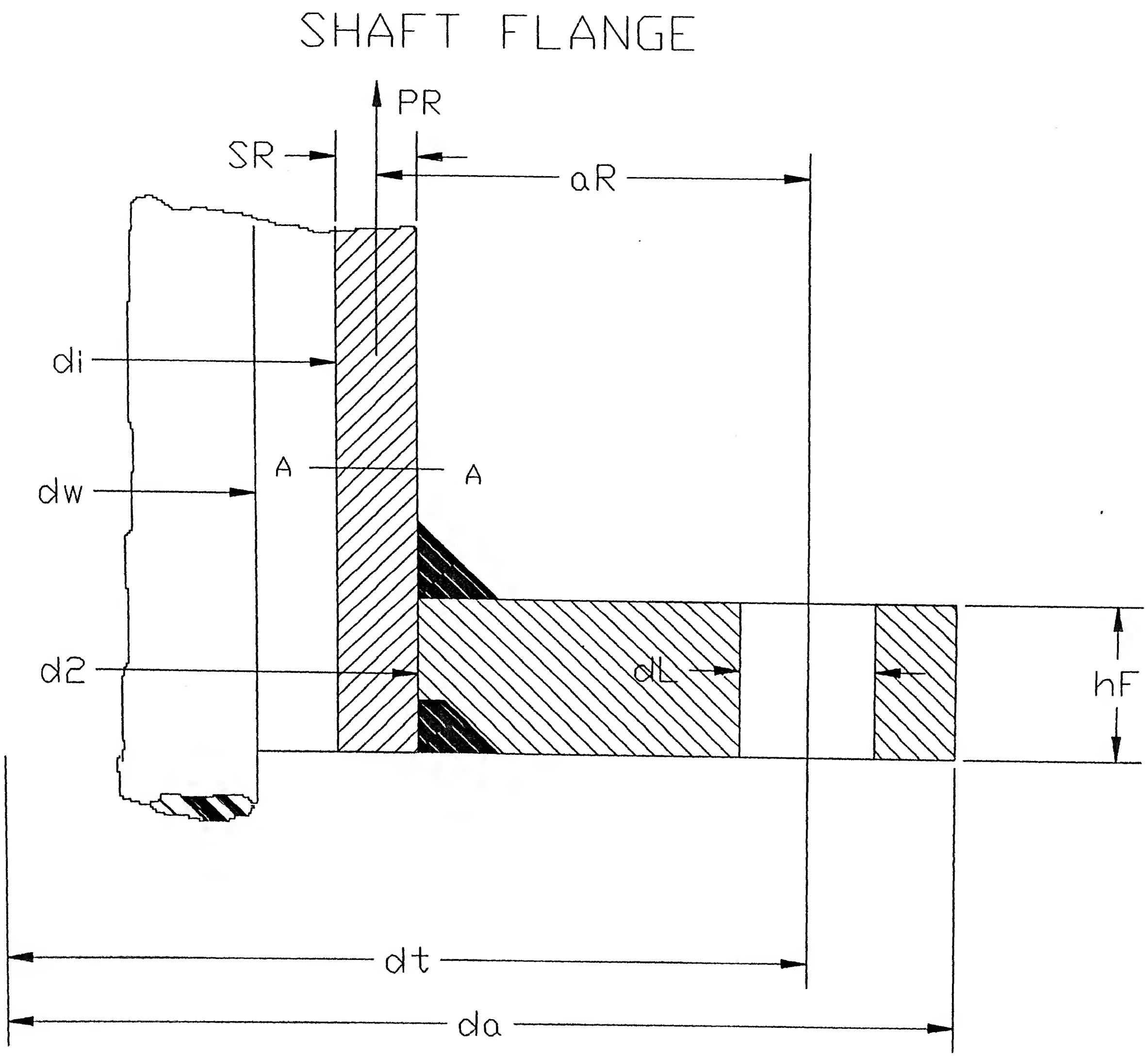


FIGURE 3.9 Sectional view of shaft flange.

DESIGN DATA:-

di (mm) ____________________Flange inner diameter.

dw(mm)______________________Shaft Diameter.

dt(mm)______________________Bolt circle diameter.

da(mm)______________________Flange outer diameter.

d2(mm)____________________ _Pipe outer diameter.

dL(mm)_____________________Dowel circle diameter.

hF(mm)_____________________Flange thickness.

SR(mm)_____________________Bearing house wall thickness.

K(MPa)_____________________Design stress value.

p1(Bar)_____________________Design pressure.

S(_)______________________Design factor of safety.

σ_{vzul}(MPa)__________________Allowable design stress.

σ_v(MPa)____________________Calculated stress in body wall through section A-A.

3.2.11 DESIGN OF COVER FOR BODY NON DRIVING END

Select following inputs for design of cover:-

a) Material for cover.

b) Design factor of safety.

Reference is made to figure 3.10

$$d_{si} = 1.2 * Shaft\ dia. \tag{3.12}$$

$$d_{so} = 1.2 * Shaft\ dia + 2.0 * Wall\ thickness.$$

$$d_D = \frac{(d_{si} + d_{so})}{2}.$$

$$s = 1.2 * d_D * \sqrt{p1.\frac{S}{10.K}}$$

COVER–N.D.E.

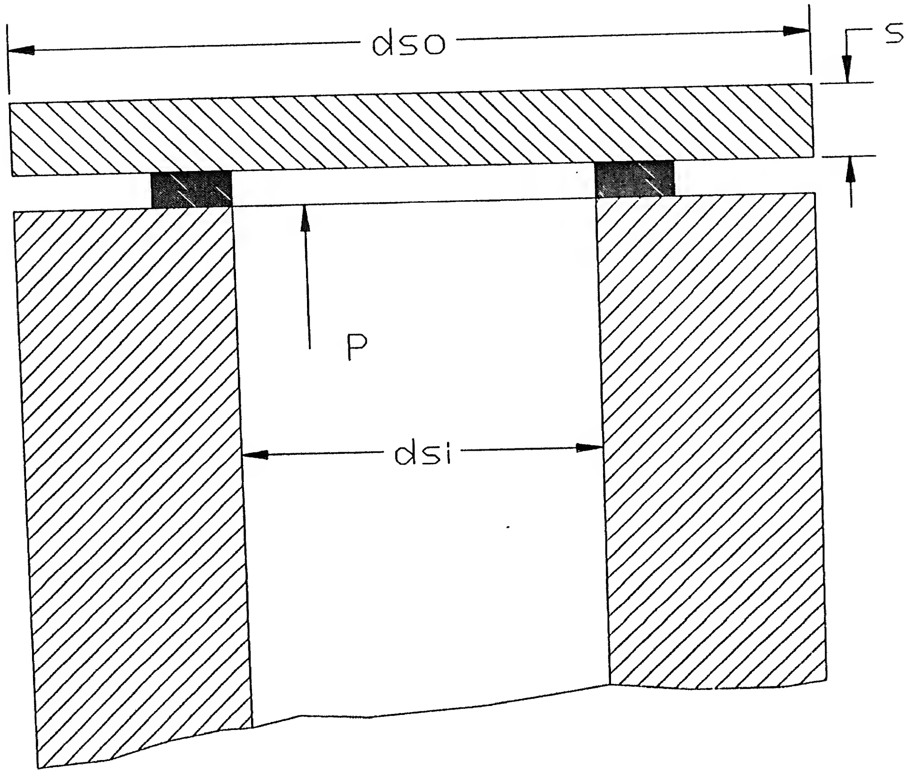


FIGURE 3.10 Sectional view of cover at non driving end.

DESIGN DATA:-

d_D ________________Diameter at location of gasket.

pi________________Design pressure.

s(_)________________Design factor of safety.

K(N/sq. mm)__________Design stress value.

3.3 SUMMARY

In this chapter, sizing of butterfly valve using nomograph technique was discussed. The design methodology of parts and their stress analysis is also discussed, and equations used are mentioned.

CHAPTER -4

DESIGN ALGORITHM AND TRIAL RUNS

4.1 INTRODUCTION

In this chapter, a stepwise procedure for the sizing of butterfly valves and design of its parts are given. The connectivity of the various segments and package implementation is attached as Appendix B, and a detail trial run of the package is at the end of this chapter. In this package, extensive use of MATLAB has been used for curve fitting of charts and expressing them as a polynomial functions, suitable for computer applications. Thus, this program stores most of the charts like NOMOGRAPH, ADAMS CHART, MOODY CHART (used to find friction factor for pipes) in a retrievable form which is other wise difficult to be read by a computer. The material data structure has also been made which helps in selection of materials for parts, the data structure can easily be changed by users in the program depending upon their requirements. These are discussed as and when they are encountered in the program.

4.2 MATERIAL DATA BASE

For designing a butterfly valve, each component of it is to be made of different type of material depending upon its requirement and usage. In the program, list of material along with its code is displayed on the screen . These material database is selected from various sources like I.S.I, B.S, AISI and some feed back from leading manufacturers[8] in this field. Display of the list of material along with its code for each part design, helps the designer for easy selection of materials and cut down the design time, instead of referring to various codes and books for it. The list of material along with its code is shown below. The code number facilitates the selection of the material by the users.

4.2.1 BUTTERFLY VALVE DISC

MATERIAL	I.S GRADE	CODE
GREY IRON	FG 15	1
	FG 20	2
	FG 35	3
SPHEROIDAL	SG 80/2	4
GRAPHITE IRON	SG 60/2	5
	SG 38/2	6
ALLOY CAST IRON	Ni-Mo.CI	7
	NITRALLOY C.I	8

4.2.2 BODY WALL THICKNESS

MATERIAL	I.S GRADE	CODE
GREY IRON	FG 15	1
	FG 20	2
	FG 25	3
	FG 30	4
	FG 35	5
	FG 40	6
	SG 80/2	7
	SG 60/2	8
ALLOY CAST	Nickel C.I	9
IRON	Ni-Mo C.I	10
	Nitraalloy C.I	11
STEEL	C 10	12

4.2.3 SHAFT DIAMETER

MATERIAL SPECIFICATION	CODE
ST.ST.BS970,431/S39/ST.ST.410	1
AISI 316/AISI 304	2

4.2.4 COLLARED BRONZE BEARING

MATERIAL SPECIFICATION	CODE
CHILLED BRONZE BUSHING	1
STEEL BUSHING WITH LIGHT LINER	2

4.2.5 KEY

MATERIAL SPECIFICATION	CODE
ST.ST.970Gr316(B.S)	1
I.S C-45	2
I.S C-50	3
I.S C-55	4
I.S C-65	5

4.2.6 SPLIT THRUST RING

MATERIAL SPECIFICATION	CODE
AISI 410	1
I.S C-40	2
I.S C-50	3
I.S C-55 Mn75	4

I.S 35Mn-2-Mn-28	5

4.2.7 DESIGN OF LOCKING CAP

MATERIAL SPECIFICATION	CODE
S.G 80/2	1
S.G 60/2	2

4.2.8 BOLTS FOR LOCKING CAP

MATERIAL SPECIFICATION	CODE
C20	1
C45	2
C60	3
St.55	4
40Crl	5

4.2.9 VALVE FLANGE .

MATERIAL SPECIFICATION	CODE
F.G.40	1
S.G42/2	2
S.G60/2	3
S.G 80/2	4
35Mn2Mo28	5

4.2.10 COVER FOR BODY NDE

MATERIAL SPECIFICATION	CODE
C10	1
C20	2
C30	3
C45	4
C50	5

4.2.11 SHAFT FLANGE STRESS

MATERIAL SPECIFICATION	CODE
C20	1
C45	2
C60	3
St.55	4
40Crl	5

4.3 CURVE FITTING FOR NOMOGRAPH

The nomographs (figure 3.1,3.2) which are used for sizing of butterfly valves give relationship between percentage flow rate and disc angle for various value of K-factors in the form of curves. These curves are converted into polynomial functions by curve fitting techniques, a facility available in MATLAB package.

To improve the accuracy for curve fitting, the complicated curves were broken into segments and then curve fitting was done separately for each curve. The polynomial functions used in the program were upto 20 decimal places to improve accuracy, however here it is shown

upto two decimal places only.

4.3.1 NOMOGRAPH (FIGURE 3.1) AS A POLYNOMIAL FUNCTION

This nomograph which represents relationship between percentage flow rate and disc angle for different values of head loss factor (K), is first converted into four polynomial function $\theta 1, \theta 4, \theta 16, \theta 64$ for K=1,4,16,64. respectively.

The nomograph is expressed as polynomial function as follows.

$$\theta 1 = 1.58e\text{-}4\ Q^3 - 2.84e\text{-}2\ Q^2 + 2.06\ Q + 6.604 \qquad (4.1)$$

$$\theta 4 = 1.18e\text{-}4\ Q^3 - 1.88e\text{-}2\ Q^2 + 1.46\ Q + 4.122.$$

$$\theta 16 = 1.90e\text{-}4\ Q^3 - 2.53e\text{-}2\ Q^2 + 1.36\ Q + 7.58e\text{-}1.$$

$$\theta 64 = 2.08e\text{-}4\ Q^3 - 2.67e\text{-}2\ Q^2 + 1.21\ Q + 5.5e\text{-}1.$$

were Q is the percentage flow rate.

The disc angle (alpha) is determined using above equations and interpolation is done for in between positions as follow.

For K = 1.

alpha = θ1.

For K = 4.

alpha= θ4.

For K = 16.

alpha =θ16.

for K= 64.

alpha = θ64.

For ($1 < K < 4$)

alpha = $\theta 1 + (\theta 1 - \theta 4) * (1 - K) / 3$.

For ($4 < K < 16$)

alpha = $\theta 4 + (\theta 4 - \theta 16) * (4 - K) / 12$.

For ($16 < K < 64$)

alpha = $\theta 16 + (\theta 16 - \theta 64) * (16 - K) / 48$.

The optimal angle is calculated for 10% and 90% of flow rate using above equations.

4.3.2 NOMOGRAPH (FIGURE 3.2) AS POLYNOMIAL FUNCTION

This Nomograph gives the relationship between butterfly valve loss coefficient (Kv) and disc opening angle, which is represented in the form of curve. MATLAB has been used to find polynomial functions using curve fitting techniques. The range of Kv is very large (i.e 0.5 to 80,000), it would have been difficult to fit a single curve which may lead to very high powers and still not be accurate, hence the curve has been broken into segments to improve accuracy. The Nomograph curve shown in figure 3.2 is expressed as polynomial functions below.

For (20,000 < = K < 80,000) (4.2)

angle = $2.77e\text{-}18\ K^4 - 5.27e\text{-}13\ K^3 + 3.44e\text{-}8\ K^2 - 9.49e\text{-}4\ K + 1.81e1$.

For (2000< = K <20,000)

angle = $-1.9e\text{-}12\ K^3 + 7.82e\text{-}8\ K^2 - 1.16e\text{-}3\ K + 1.72e01$.

For (400 < = K < 2000)

angle = $-3.31e\text{-}9\ K^3 + 1.382e\text{-}5\ K^2 - 1.98e\text{-}2K + 2.48e01$.

For (30< = K < 400)

angle = $-2.219e\text{-}6\ K^3 + 1.45e\text{-}3K^2 - 2.83e\text{-}1K + 4.2e1$.

For(0.5 < = K < 3.0)

angle = $1.12e1\ K^3 + 6.5e1\ K^2 - 1.18e2\ K + 1.34\ e2$.

4.4 CALCULATION OF K-FACTOR FOR PIPES

The head loss in the pipe lines is given by

$H_L = K\ V^2 / 2g$ (4.3)

Where $K = f\ L / D$

Where f is the friction factor for pipe, D is the pipe diameter, and L is the length.

For this determination of friction factor (f) is necessary, which is expressed in the form of Moody's chart (figure 4.1). This chart has been expressed in the form of equation for computer to store Moody chart in retrievable form which is discussed below.

Select material of pipe as displayed on screen, which makes program to select absolute roughness (e) in meters.

MATERIALS FOR PIPE	ABSOLUTE ROUGHNESS (e) IN METERS
a) Glass, new commercial pipe surface drawn tubing(brass,copper,lead)	1.52 e-6

b) Commercial steel or wrought iron	4.57e-5
c) Asphalted cast iron	1.22e-4
d) Galvanized	1.52e-4
e) cast iron	2.59e-4
f) Wood stove	2.59e-4
g) Concrete	1.83e-4 to 9.14e-5
h) Riveted steel	3.05e-4 to 3.05e-3

Calculate Reynold number (Re)

$Re = \rho \ v \ d \ / \ \mu.$

Where μ is the kinematic viscosity and ρ is the density of the fluid.

Calculate friction factor (f)

For $Re > 10^4$ i.e turbulent and $10^{-5} < G < 0.04$

were $G = e/d$

$$f = a + b\ Re^{-c} \tag{4.4}$$

$a = 0.094G^{0.225} + 0.53\ G.$

$b = 88\ G^{0.44}$

$c=1.62\ G^{0.134}$

Once friction factor(f) is known, K factor is determine from equation (4.3)

4.5 ADAM'S CHART AS POLYNOMIAL FUNCTIONS

The ADAM'S chart attached as Appendix C, are first converted into graph and then their polynomial functions by curve fitting technique of MATLAB.

The ADAM's CHART is used to find.

a) Shaft diameter as per valve size.

b) Seat torque as per valve size.

c) Valve offset as per valve size.

4.5.1 SHAFT DIAMETER (d)

Reference is made to figure 4.1, 4.2. This figure gives relationship between shaft diameter and

valve size as per material selected.

-For material : St.St.B.S 970,431 s/29 St.St.410 heat treated.

For ($P > 0.0$ and $P <= 6.0$)

$d=1.072e\text{-}8\ D^3 - 4.5e\text{-}5\ D^2 + 0.13D - 0.6856.$

For ($P > 6.0$ and $P <= 16$)

$d = 3.59e\text{-}9\ D^2 + 0.08\ D + 25.58$

For ($P > 10$ and $P <= 16$)

$d= -0.0001D^2 + 0.3667\ D - 170.0.$

were P is the pressure in Bar and D is the disc diameter in mm.

-For material :AISI 316 / AISI 304

For ($P>0.0$ and $P <6.0$)

$d = 1.99e\text{-}8\ D^3 - 6.86e\text{-}5\ D^2 + 1.66e\text{-}1D + 4.71e\text{-}1.$

For ($P >6.0$ and $P<=10.0$)

$d= 1.36e\text{-}5D^2 + 1.64e\text{-}1D - 4.95e\text{-}1.$

For ($P >10.0$ and $P<=16$)

$d= 1.77e\text{-}5\ D^2 + 1.45e\text{-}1\ D - 3.27e\text{-}2.$

4.5.2 SEAT TORQUE

Reference is made to figure 4.4. This gives the relation ship between seat torque and valve size.

Seat torque = $2.207e\text{-}6D^3 - 8.57e\text{-}4D^2 + 2.65D - 2.37e2.$

4.5.3 VALVE OFFSET

Offset = $1.72e\text{-}9\ D^3 - 7.356e\text{-}6\ D^2 + 0.102D + 0.1825.$

In all the above cases, in finding the polynomial functions by curve fitting techniques by MATLAB, an error was observed in the range of 2 to 8.5 percent, even after taking accuracy of twenty decimal places, and breaking the curve into small segments. This error is neglible as far as this package is concerned, because valves are made in discreet size (10,15, 25 ......) due to availability of off the shaft flanges in standard sizes in the market, thus the calculated valve size is always rounded off to next higher size to meet the flange size available in the market. The shaft diameter calculated by ADAMS CHART always under go stress analysis, where proper factor of safety is buit in. Hence any small error in this case can also be ignored.

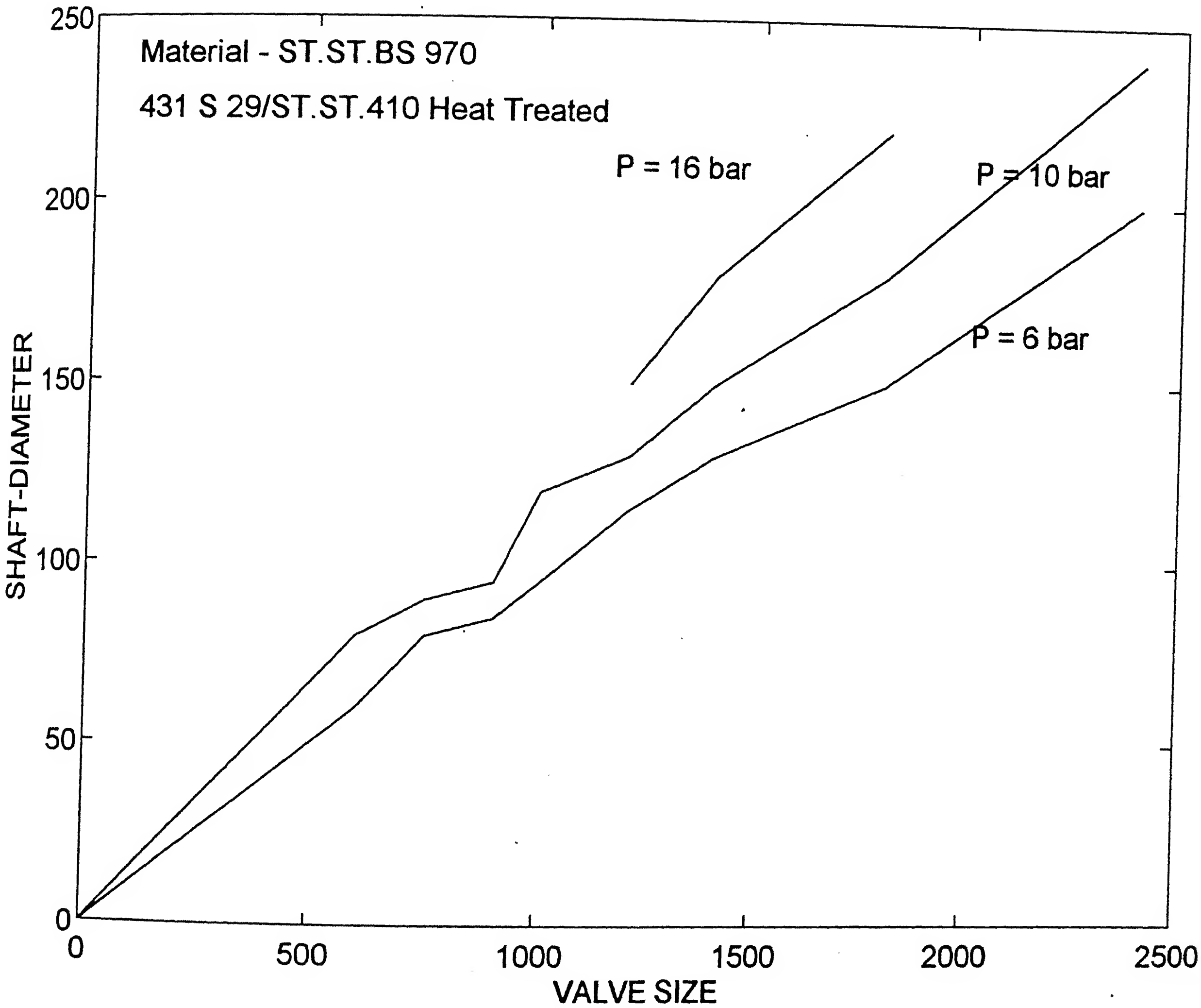


FIGURE 4.1 Relation ship of shaft diameter to valve disc at different pressure.
(MATERIAL - ST.ST.BS 970,431S 29/ST.ST.410 HEAT TREATED)

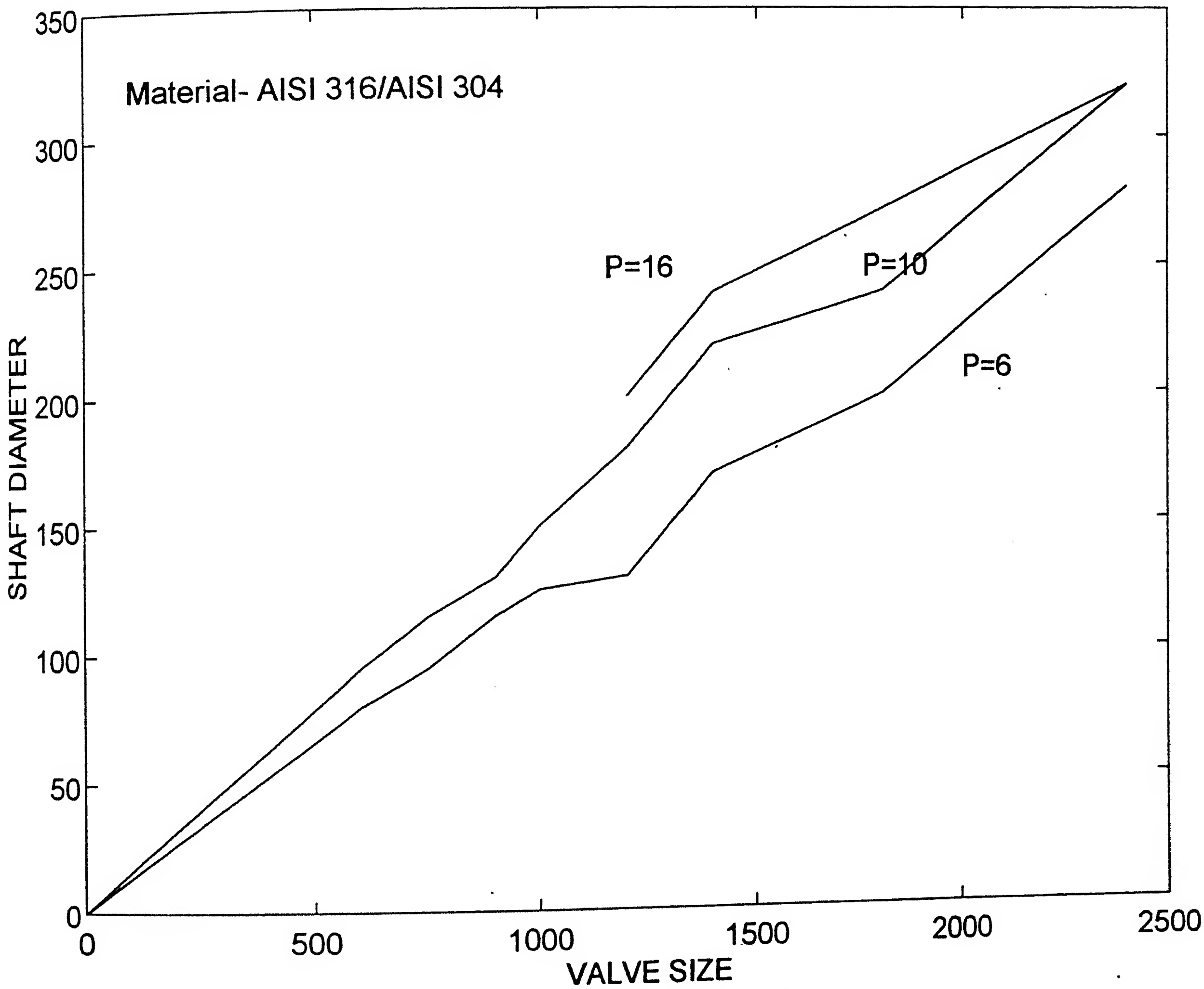


FIGURE 4.2 Relation ship of shaft diameter to valve disc at different pressure.
(MATERIAL- AISI 316 / AISI 304)

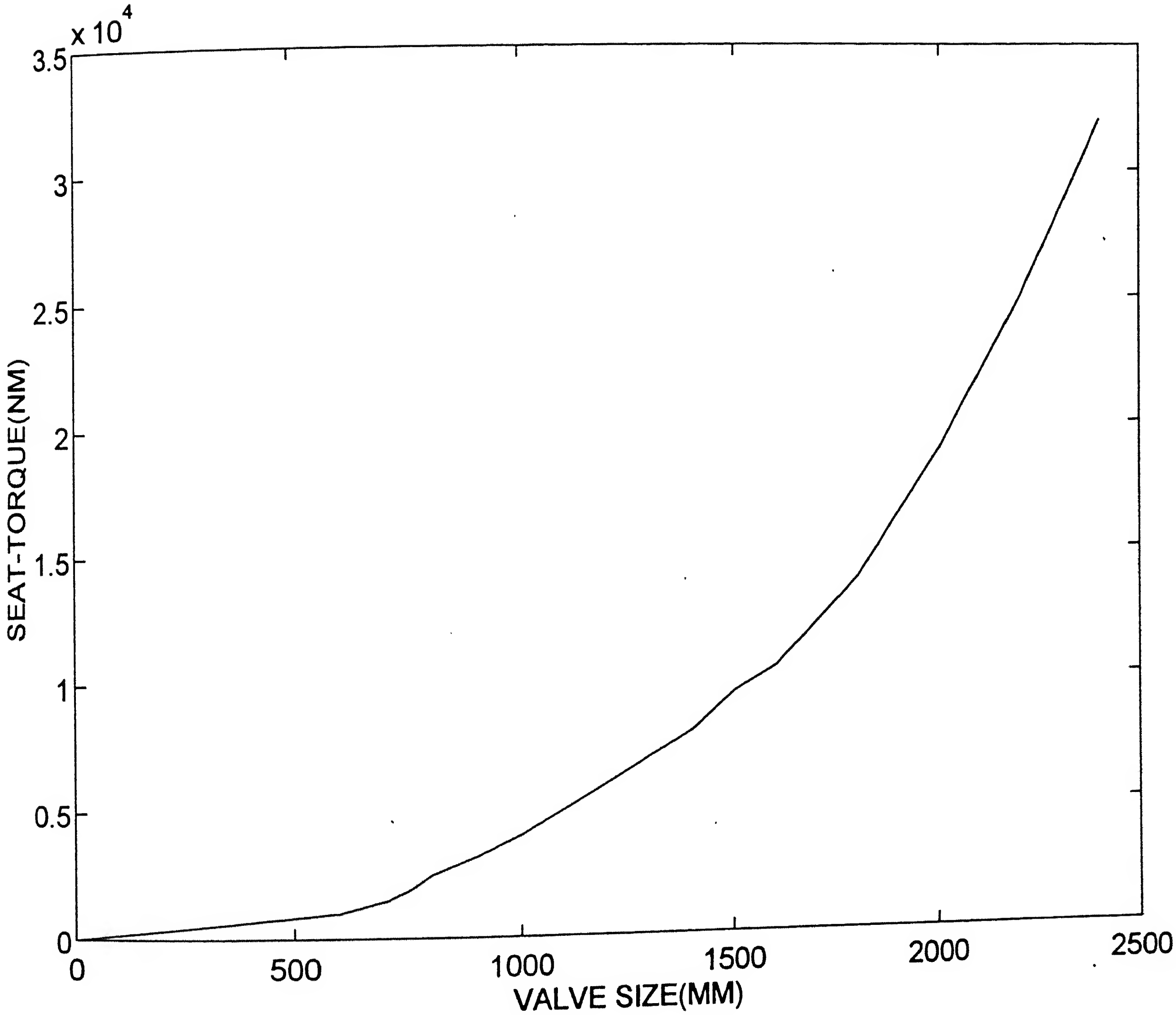


FIGURE 4.3 Relationship of seat torque to valve disc diameter.

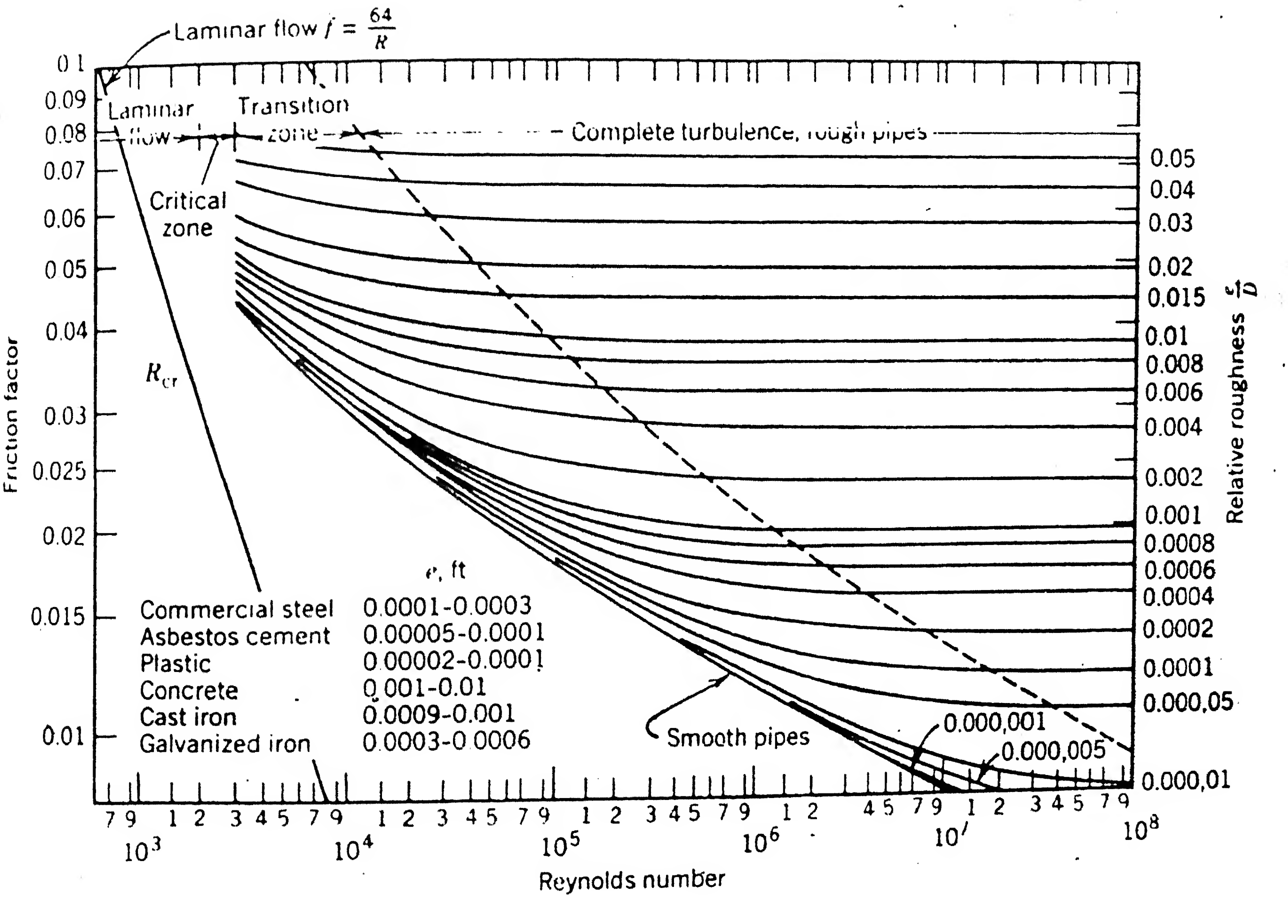


Figure 4.4 Moody chart

4.6 DESIGN ALGORITHMS FOR PARTS / STRESS ANALYSIS

Design algorithms for various parts / stress analysis of butterfly valve is shown below step by step:-

4.6.1 CALCULATION OF DISC THICKNESS

.Interact with users to select material as displayed on the screen.

Input design factor of safety.

Calculate thickness and maximum deflection from equation(3.1), and display the results on the screen.

Display sectional views and results in C-graphic mode.

4.6.2 STRESS ANALYSIS OF DISC

Design disc of lens shape .

Carry out stress analysis of disc from equation (3.2).

Display shape of designed disc with results in graphic mode.

If stress exceeds the design stress display warning message.

4.6.3 BODY WALL THICKNESS

Interact with users to select material as displayed on the screen.

Input design factor of safety.

Calculate wall thickness from equation (3.3) and display results on screen.

Display sectional views and results on screen in graphic mode.

4.6.4 CALCULATION OF SHAFT DIAMETER , SEAT TORQUE, VALVE OFFSET AND TOTAL TORQUE ACTING ON SHAFT

Calculate shaft diameter as per valve size from ADAM'S CHART expressed as polynomial function .

Calculate seat torque as per valve size from ADAM'S CHART expressed a polynomial function.

Calculate valve offset as per valve size from ADAM'S CHART expressed as polynomial function.

Calculate bearing torque, offset torque from equation(3.4), and the total torque acting on shaft.

Total torque = Seat torque + bearing torque + offset torque.

Display results.

Display sectional view of shaft with results in graphic mode.

4.6.5 DESIGN OF COLLARED BRONZE BEARING

Interact with users to select material as displayed on the screen.

Input design factor of safety.

Calculate dimensions of bearing from equation(3.6) and display results

Display sectional view of designed bearing along with results in graphic mode.

4.6.6 STRESS ANALYSIS OF SHAFT

Calculate stresses developed on the shaft due to combined bending and torsion effect from equation (3.6)..

If stresses developed in the shaft is more than design stress, display warning message on the screen.

display results on the screen.

4.6.7 DESIGN OF SHAFT KEY / STRESS ANALYSIS

Interact with users to select material as displayed on the screen.

Input design factor of safety.

Select numbers of keys required.

Calculate dimensions of key from equation (3.7).

Calculate
a) Surface stress developed in shaft.
b) Surface stress developed in hub.
c) Induced stresses.

Display warning if stresses developed in shaft exceeds the design stress.

Display sectional view along with results in graphic mode.

4.6.8 DESIGN OF SPLIT THRUST RING / STRESS ANALYSIS

Interact with users to select material as displayed on the screen.

Input design factor of safety.

Calculate dimensions from equation (3.8).

From equation (3.8), find
a) Bearing stresses.
b) Bending stresses.

If stresses in shaft exceeds the design stress display warning message.

Display sectional views along with results in graphic mode.

4.6.9 DESIGN OF LOCKING CAP

Interact with users to select material as displayed on the screen.

.Input design factor of safety.

Calculate dimensions of locking cap from equation (3.9a).

4.6.10 DESIGN OF BOLTS FOR LOCKING CAP

a Interact with users to select material as displayed on the screen.

b) Input design factor of safety.

c) Select numbers of bolts.

Calculate core area and core diameter of bolts from equation (3.9b).

Display sectional view of locking cap along with bolts and results in graphic mode.

4.6.11 VALVE FLANGE STRESS ANALYSIS

.Interact with users to select material as displayed on the screen.

Input design factor of safety.

Input standard flange dimensions as per valve size from I.S : 6392-1971.

Carry out stress analysis from equation (3.10).

If stresses developed in sections > design stress value, display warning message.

Display sectional view along with results in graphic mode.

4.6.12 FLANGE GASKET DIMENSIONS

Calculate and display dimensions of flange gasket dimensions.

4.6.13 DESIGN OF COVER FOR BODY N.D.E

Interact with users to select material as displayed on the screen.

Input design factor of safety.

Calculate and display dimensions of cover from equation (3. 12).

4.6.14 SHAFT FLANGE STRESS ANALYSIS

Interact with users to select material as displayed on the screen.

Input design factor of safety.

Input standard flange dimensions as per shaft size from I.S.I specifications IS: 6392-1971.

Carry out stress analysis as per form equation (3.11).

If stresses developed in sections > design stress value, display warning message.

Display sectional view along with results in graphic mode.

4.6.15 TORQUES AND PRESSURE DROP AT VARIOUS DISC ANGLE

Calculate following variables at various disc angle(0 to 90 degrees).

a) Head loss factor(K).

b) Pressure drop

c) Bearing torques

d) Offset torques

e) Hydraulic torque

f) Torques for closing

g) Torques for opening.

Display above results in the form of chart on screen.

Display line graph between various disc position and pressure drop.

Display the bar chart between various disc position and pressure drop.

4.6.16 PREPARATION OF SUMMARY OF RESULTS

Open file valve.out and write summary of results.

4.6.17 PREPARATION OF SCRIPT FILE

Open file valve.scr and write script file, which when loaded AUTOCAD prepare final drawing along with bills of materials.

4.7 **ILLUSTRATIVE DESIGN AND TRIAL RUN**

To design a butterfly valve for Hydro-power plant with following specifications:

1) Type of valve	Butterfly valve.
2) Maximum flow rate required	15,000 cubic meters / hour.
3) Minimum flow rate required	12,000 cubic meters / hour.
4) Total available head	100 m.
5) Maximum pressure	8 bar.
6) Diameter of pipe line	2000 mm
7) Total length of pipe	200m.
8) Material of pipe	Concrete.
9) Number of bend	2.

10) Type of entrance Dull edge.

Select valve flange dimensions as per IS:6392-1971, Assumption of any unspecified conditions and selection of materials for parts is at designer discretion.

4.7.1 **RESULTS:-**

The program first does the sizing of valve and the optimum valve size obtained for above conditions is 1587.23 mm, since the flanges are available in the market in discret size as per IS:1703-1968 , we select valve size of 1600 mm for designing its parts.

The out put of the program is stored in a seperate file "valve.out" which consist of major input parameters followed by output obtained. The sectional views of parts is obtained in C-Graphic mode, some photographs of the screen output are shown in figure 4.5 to 4.16.

The final drawing along with BILLS OF MATERIAL is in AUTOCAD (figure 4.17) The out put of file" valve.out " is shown below:-

4.6.2 INPUT PARAMETERS

Disc diameter	1600mm
Operating pressure	8bar
Maximum flow rate	15,000 m^3 / hour.
Minimum flow rate	12,000 m^3 / hour

4.6.3 OUTPUT DATA

VALVE DISC THICKNESS

Selected material	GREY IRON FG 35.
Thickness of disc	51.78 mm
Maximum deflection	9.44242e-05 mm

DISC SHAPE PARAMETERS AND STRESS ANALYSIS

With reference to figure 3.1

Alpha	53.79 degrees
Beta	58.86 degrees
Value of	
R	1440.0 mm
E	216.0 mm

C	1120 mm
S_S	34.5 m
Stress developed in disc	96.06 N/sq mm.

BODY WALL THICKNESS

Selected material	FG20
Wall thickness	76.51 mm
Outer diameter	1676.51 mm

DESIGN OF SHAFT, OFFSET AND TORQUES

Material specification	St.ST.B970, 431/S39/ST.ST.410.
Shaft diameter	164.00 mm
Offset	5.0mm
Seat torque	10859.68 N-m
Bearing torque	20180.18 N-m
Offset torque	8038.40 N-m
Toal torque	39078.27 N-m

DESIGN OF COLLARED BEARING

Material	Steel bushing with light liner
Length	140.041 mm
Clearence	0.328 mm
Thickness of bush	13.42 mm
Collar diameter	196.8 mm

STRESS ANALYSIS OF SHAFT

Stress developed in shaft	141.09 N / mm^2

DESIGN OF SUNK KEY OF RECTANGULAR CROSS-SECTION

Material selected	I.S C-50
Number of keys selected	2
Length	265.5 mm
Bredth	44.25 mm
Height	29.5 mm

STRESS ANALYSIS OF KEY

Surface stress in shaft	80.68 N /sq mm

Surface stress in hub	66.54 N /Sq mm
Induced stress	20.28 N/sq mm

DESIGN AND STRESS ANALYSIS OF SPLIT THRUST RING

Material specification	I.S C-40
Outer diameter	196.79 mm
Inner diameter	147.59 mm
Bredth	8.2 mm
Bearing Stress	42.11 kg /sqcm
Design stress	266.67 kg / sq cm

DESIGN OF LOCKING CAP

Material specification	I.S 80 /2
d	164.0mm
D1	205.0 mm
D2	492.0 mm
Thickness	1.97 mm
Pitch circle diameter	348.5 mm

DESIGN OF BOLTS FOR LOCKING CAP

Number of bolts	6
Core area	786.18 sq mm
Core diameter	31.64 mm

BUTTERFLY VALVE FLANGE STRESS ANALYSIS

Material selected	35 Mn 2 Mo28
Stress developed in section A-A	120.03N / sq mm
Stress developed in section B-B	151.14 N /sq mm
Stress developed in section C-C	59.68 N /sq mm

DIMENTIONS FOR FLANGE GASKET (SOFT MATERIAL,SOFT METAL OR FLAT METAL TYPE)

Inner diameter	1620.0 mm
Outer diameter	1750.0 mm
Thickness	31.82 mm

DESIGN OF COVER FOR BODY NON DRIVING END

Material selecteu

Thickness of cover(t) 31.82 mm

DIMENTIONS FOR GASKET FOR COVER

Gasket inner diameter 196.80 mm

Gasket outer diameter 349.82 mm

SHAFT FLANGE STRESS ANALYSIS

Stress developed in section A-A 96.49 N /sq mm

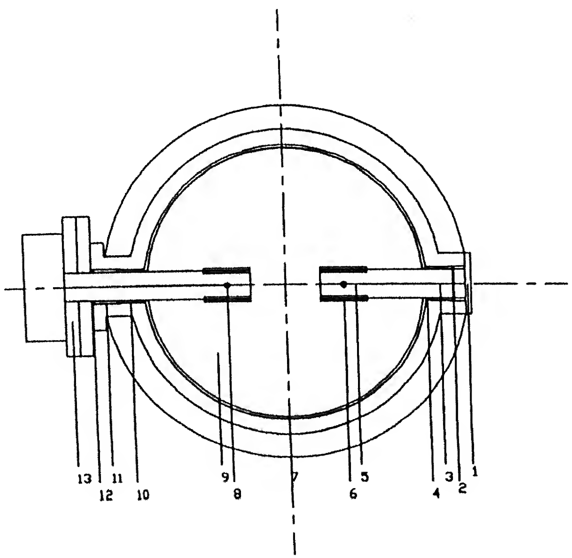
PRESSURE DROP AND TORQUES AT VARIOUS DISC POSITIONS

Angle	K	Pressure Drop	Bearing Torque	Offset Torque	Hyd Torque	Torque Closing	Torque Opening
0.0	1000.0	8.0	20170.0	8038.4	32768.0	60976.4	-4559.6
10.0	211.2	4.52	11414.9	4549.2	18544.6	34508.8	-2580.5
30.0	23.5	0.50	1270.0	506.1	2063.3	3839.4	-287.1
30.0	7.4	0.15	398.1	158.7	646.8	1203.6	-90.0
40.0	4.4	0.09	239.6	95.5	389.2	724.3	-54.2
50.0	1.5	0.03	81.0	32.3	131.6	244.9	-18.3
60.0	1.0	0.02	55.7	22.2	90.5	168.4	-12.6
70.0	0.6	0.01	30.4	12.1	49.4	92.0	-6.9
80.0	0.4	0.007	19.0	7.6	30.9	57.6	-4.3
90.0	0.2	0.005	13.1	5.2	21.2	39.5	-3.0

4.8 SUMMARY

In this chapter collection of material data base, and procedure of converting Nomograph, Adams chart, Moody chart into polynomial function was given. The design algorithms for parts and stress analysis are also given, finally an illustrative design along with trial run was discussed.

CROSS-SECTIONAL VIEW OF BUTTERFLY VALVE



Sr.No	DESCRIPTION	MATERIAL
1.	COVER FOR BODY N.D.E	C-10
2.	SPLIT THRUST RING	I.S_C-40
3.	FLANGED BUSH	STEEL_BUSHING_WITH_LIGHT_L
4.	PACKING FOR BUSH	GRAPHITED ASBESTOS
5.	DRIVE SHAFT	ST.ST.BS970,431/S39/ST.ST.410
6.	KEY	I.S_C-50
7.	FLANGE	35_Mn2Mo28
8.	DOWEL PIN	ST.ST.BS.970.Gr316S16
9.	DISC	GREY_IRON_FG_35
10.	VALVE BODY	FG-20
11.	SPACER RING	M.S
12	'U'CUP SEAL	BRONZE I.S-318 Gr2
13.	ADAPTER PLATE	M.S I.S226 GrFe410S

Figure 4.17 Final drawing in AUTOCAD

CHAPTER-5

CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER WORK

5.1 CONCLUSIONS

The present work is an attempt to develop an interactive design package for butterfly valves. This package will be very useful for irrigation department , Hydro powers plant and industries like Kirloskar Brothers which manufactures these type of valves for commercial purposes. This package can also be used for educational aid since the program is very interactive, easy to operate and display graphic views at every stage of design for easy understanding.

In practice, the entire process of designing a butterfly valves consist of sizing of butterfly valve, which consist of number of iterations and look up charts, and designing its parts with material selection combinations. When this process is carried out manually, it is difficult to explore all the alternative strategies and consumes many man-hours. The job becomes monotonous if iterations is carried out manually, since each iteration is very elaborate and time consuming.

The salient feature of this package is that designer can quickly explore all the alternative design strategies with various material selection combinations and select appropriate design. For this purpose, at every design stage program interact with users and display sectional views of the designed part in graphic mode to give an actual feel of designing.

5.2 RECOMMENDATIONS FOR FURTHER WORK

The present version of the program does not deals with the practical Seal design and seats, which is a separate topic in itself, if butterfly valves is to be designed for Petrochemical industries, Oil refineries where highly inflammable,toxic liquids and gases is to handled, and were seepage of fluids can not be tolerated. The design of actuators like hand wheel, electric motors limit switches, torque switches and gear box should be taken into account to avoid loadin operation and stopping at intermittent position other than fully open and closed position.

It is hoped that enhanced version of the design package should take into account th above mentioned points.

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USER'S MANUAL

This manual describes the usage of this package and the sequence of events, prompts appearing on the screen. Prompts (:) are self explanatory however, a brief description is given wherever necessary.

Run the program by loading batch file vajra.bat.
(type "vajra" at prompt sign)
Hit any key to flip between text and graphic mode anywhere in the program.

1. Do you want to do sizing of valve (Y / N)
(Enter yes if you want to do sizing else enter no)
SIZING OF VALVE
2. a) Input max flow rate required (cubic meter / hour)
    b) Input min flow rate required (cubic meter / hour)
    c) Input max operating pressure
3.a) Input pipe diameter (in mm)
    b) Input total available head (in meter)
    c) Select code for type of entrance
      (List of type of entrance is displayed on the screen)
    d) Enter number of bends
    e .Input angle of bends in degrees
    f) Input total length of pipe(in meters)
    g) Select code for pipe material
      (list of material is displayed)
4 Program displays optimum valve size

DISC THICKNESS/ STRESS ANALYSIS

5. a) Select material code for disc

(List of material along with codes is displayed)

b) Enter factor of safety

6.a) Display sectional view and results in graphic mode.

b) Do you want to redesign the disc(Y/N) :

BODY WALL THICKNESS

7. a) Select material code for wall :

(List of material along with codes is displayed)

b) Enter factor of safety :

8. Display sectional view and results in graphic mode.

SHAFT DIAMETER AND TORQUES

9 a) Select material code for shaft :

(List of material along with codes is displayed)

b) Enter factor of safety :

10. Display sectional view and results in graphic mode.

COLLARED BRONZE BEARING

11. a) Select material code for bearing:

(List of material along with codes is displayed)

b) Enter factor of safety :

12. Display sectional view and results in graphic mode.

STRESS ANALYSIS OF SHAFT

13. Display results.

DESIGN OF SHAFT KEY / STRESS ANALYSIS

14. a) Select material code for key

(List of material along with codes is displayed)

b) Enter factor of safety

c) Enter numbers of key required

15 Display sectional view and results in graphic mode.

.DESIGN OF SPLIT THRUST RING

16. a) Select material code for split thrust ring

(List of material along with codes is displayed)

b) Enter factor of safety .

17 Display sectional view and results in graphic mode.

DESIGN OF LOCKING CAP

18. a) Select material code for locking cap.

(List of material along with codes is displayed)

b) Enter factor of safety :

19.Display sectional view and results in graphic mode.

DESIGN OF BOLTS FOR LOCKING CAP

20. a) Select material code for bolts:

(List of material along with codes is displayed)

b) Enter factor of safety :

c) Select numbers of bolts.

21.Display sectional view and results in graphic mode.

VALVE FLANGE STRESS ANALYSIS

22. a) Select material code for valve flange:

(List of material along with codes is displayed)

b) Enter factor of safety :

23 Input standard flange dimensions as per valve size from IS: 6391-1971

24. Display sectional view and results in graphic mode.

FLANGE GASKET DESIGN

25. a .Select material code for gasket:

(List of material along with codes is displayed)

b) Enter factor of safety :

26.Display sectional view and results in graphic mode.

DESIGN OF COVER FOR BODY N.D.E

27. a) Select material code for cover

(List of material along with codes is displayed)

b) Enter factor of safety -

28 Display sectional view and results in graphic mode.

SHAFT FLANGE STRESS ANALYSIS

29. a) Select material code for flange:

(List of material along with codes is displayed)

b) Enter factor of safety :

30 Input standard flange dimensions as per valve size from IS:6392-1971.

31 Display sectional view and results in graphic mode.

32 Displays torques and pressure drop at various disc positions.

32 Open "valve.out " file to see summary of results at the end of program.

33 Open AUTOCAD and load script file "draw" , to see final assembled drawing of butterfly valve along with bills of materials.

Appendix-B

DESIGN PROCEDURE FOR PACKAGE AND IMPLEMENTATION

The package BUTTERFLY has been coded in C-Language with an extensive use of C-Graphics and Autocad. The whole program is split up into five files of 800-1000 lines each and their executable files are clubbed into one batch file named BUTTERFLY.BAT for continuity of program and to avoid out of memory problems in computer. Each file in itself contains a main program which than calls users written functions to perform specific task. At every design stage program interact with user and display sectional-views of the part with results in C-Graphics to give an actual feel designing. Finally it draws the assembled view of valve in Autocad along with BILLS OF MATERIAL.

The design algorithm is discussed below

The Batch file BUTTERFLY.BAT contains the following executable files:

a) Valve1.exe.
b) Valve2.exe
c) Valve3.exe.
d) Valve4.exe.
e) Valve5.exe.

The program is discussed below file by file.

FILE 1 : VALVE 1.C

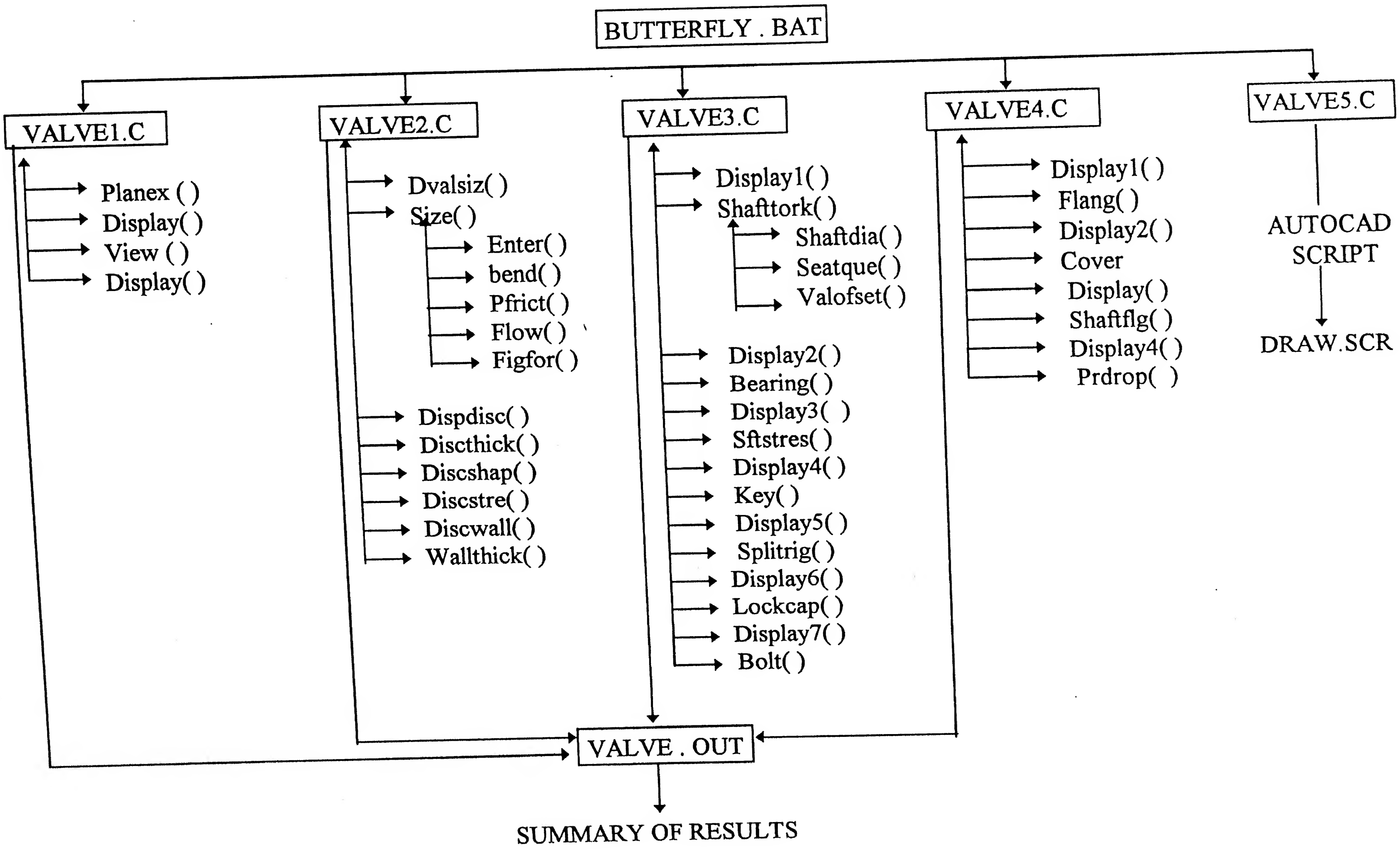
PURPOSE :- 1) To display and animate the main title "COMPUTER AIDED DESIGN OF BUTTERFLY VALVE"in graphics and in text mode.

2) To display sectional view of butterfly valve and to animate the movement of disc in C-Graphic mode to give actual feel to user about valve before design.

FILE 2: VALVE 2.C

This program interact with users for basic input data, do valve sizing, design parts ,do stress analysis and display them in graphic mode along with results. The main program calls the following functions in steps to perform the task.

STEPS	FUNCTIONS	PURPOSE
1.	dvalsiz( )	To display title"VALVE SIZING" in graphic mode for 10 sec.
2.	size( )	Do sizing of valve by calling following sub functions:
2.a	enter( )	Calculate the entrance losses in pipe.
2.b	bend( )	Calculate bend losses in pipe.
2.c	pfrict( )	Calculate frictional losses in pipe.
2.d	flow( )	Calculate optimal control angle.
2.e	figfor( )	Read values of angles from nomograph.
3	dispdisc( )	Display"VALVE DISC THICKNESS" in graphic mode for 10 sec.
4.	discthick( )	Calculate thickness of disc and display sectional view in graphic mode.
5.	discshap( )	Display shape of disc in graphic mode .
6.	discstre( )	Carry out stress analysis of disc.
7.	dispwall( )	Display"VALVE BODY THICKNESS" in graphic mode.
8.	wallthick( )	Calculate thickness of valve and display sectional view in graphic mode.


BUTTERFLY . BAT
VALVE1.C
Planex ( )
Display( )
View ( )
Display( )
VALVE2.C
Dvalsiz( )
Size( )
Enter( )
bend( )
Pfrict( )
Flow( )
Figfor( )
Dispdisc( )
Discthick( )
Discshap( )
Discstre( )
Discwall( )
Wallthick( )
VALVE3.C
Display1( )
Shafttork( )
Shaftdia( )
Seatque( )
Valofset( )
Display2( )
Bearing( )
Display3( )
Sftstres( )
Display4( )
Key( )
Display5( )
Splitrig( )
Display6( )
Lockcap( )
Display7( )
Bolt( )
VALVE4.C
Display1( )
Flang( )
Display2( )
Cover
Display( )
Shaftflg( )
Display4( )
Prdrop( )
VALVE5.C
AUTOCAD
SCRIPT
DRAW.SCR
VALVE . OUT
SUMMARY OF RESULTS

FILE 3: VALVE 3.C

This program interact with user for input data and design parts of valve, do stress analysis and display sectional views of parts in graphic mode. The main program calls the following functions in steps to perform task :

STEP	FUNCTIONS	PURPOSE
1.	display1( )	Display ".DESIGN AND STRESS ANALYSIS OF VALVE DRIVE SHAFT" in graphic mode.
2.	shaftork( )	Calculate total torque acting on shaft by calling following sub-functions:
2a.	shaftdia( )	Read shaft diameter as per valve size from ADAM'S CHART.
2b.	seatque( )	Read seat torque as per valve size from ADAM'S CHART
2c.	valofset( )	Read valve offset as per valve size from ADAM'S CHART

STEP	FUNCTION	PURPOSE
3.	display2( )	Display "DESIGN OF JOURNAL BEARING"in graphic mode.
4.	bearing( )	Design bearing and display sectional view in graphic mode.
5.	display3( )	Display "DRIVE SHAFT STRESS ANALYSIS IN GRAPHIC MODE"in graphic mode.
6.	sftstres( )	Do stress analysis of shaft for combined bending and torsion.

7.	display4( )	Display"DESIGN OF SUNK KEY OF RECTANGULAR SECTION".
8.	key( )	Design key,carry out stress analysis and display sectional view in graphic mode.
9.	display5( )	Display "DESIGN OF SPLIT THRUST RING".in graphic mode.
10.	splitrig( )	Design split thrust ring and display sectional view in graphic mode.
11	display6.( )	Display"DESIGN OF LOCKING CAP".
12.	lockcap( )	Design locking cap and display sectional view in graphic mode.
13.	display7.( )	Display"DESIGN OF BOLTS".
14.	bolt( )	Design bolt for locking cap.

FILE 4:-VALVE 4.C

This program carry out stress analysis of valve/shaft flange,calculate K-factor,pressure drop,hydraulic torque ,torque for closing, torque for opening etc at various disc position and display the graph between pressure drop and disc angle in C-Graphics.It call the following function to perform the task.

STEP	FUNCTIONS	PURPOSE
1.	display1 ( )	Display"VALVE FLANGE STRESS ANALYSIS" in graphic mode.
2.	flang( )	Carry out stress analysis of valve flange and display sectional view in graphic mode.
3.	display2( )	display"COVER FOR N.D.E".

STEP	FUNCTIONS	PURPOSE
4.	cover( )	Design cover for N.D.E.
5.	display3 ( )	Display"SHAFT FLANG STRESS ANALYSIS".
6.	shaftflg( )	Carry out stress analysis for shaft flange.
7.	display4( )	Display"PRESSURE DROP Vs DISC ANGLE".
8.	prdrop( )	a.)Calculate K,pressure drop,total torque,hydraulic torque,torque for closing,torque for opening at various disc position. b)Display line graph between disc angle and pressure drop. c) Display bar graph between disc angle and pressure drop in C-Graphics.

FILE 5: VALVE 5.C

PURPOSE: This program read design data from the first four file and make script file "DRAW.SCR" for Autocad.Loading "DRAW.SCR" in Autocad draws final assembled drawing of butterfly valve along with BILLS OF MATERIAL.

FILE: VALVE.OUT

PURPOSE: This file contains the summary of results of final design of butterfly valve,which can be seen on terminal at any time.

TABLE -1

CHART OF OFFSET FOR VARIOUS SIZES OF BUTTERFLY VALVES

SR.NO.	VALVE SIZE (M M)	OFF-SET (M M)
1)	450 - 1000	04
2)	1200 -1800	05
3)	2400	06

TABLE -2 SEAT TORQUE (NM)

SIZE MM	SEAT TORQUE (NM) 2.5 BAR	6 BAR	10 BAR	16 BAR	25 BAR
600	--	1000	1000	1000	1000
700	--	1500	1500	1500	1500
750	--	1900	1900	1900	1900
800	--	2500	2500	2500	2500
900	--	3200	3200	3200	---
1000	--	4000	4000	4000	---
1100	--	5000	5000	5000	---
1200	6000	6000	6000	6000	---
1300	7000	7000	7000	---	---
1400	8000	8000	8000	---	---
1500	9500	9500	9500	---	----
1600	10500	10500	10500	----	----
1800	14000	14000	-----	----	----
2000	19000	19000	----	-----	-----
2200	25000	25000	-----	-----	-----
2400	32000	32000	-----	------	-----

NOTE : Above Torque Values are substracted from k138 chart of ADAMS

TABLE 2.1 SHAFT DIA.

MATERAL -ST.ST.BS 970 ,431 S 29/ ST.ST.410 HEAT TREATED

VALVE SIZE	SHAFT DIA.(mm)		
MM	6 BAR	10 BAR	16 BAR
600	60	80	--
750	80	80	--
900	80	95	--
1000	95	--	--
1200	115	130	150
1400	--	150	180
1800	150	180	220
2400	200	240	--

TABLE 2.2 SHAFT DIA.

MATERAL - AISI 316/AISI 304

VALVE SIZE	SHAFT DIA.(mm)		
MM	6 BAR	10 BAR	16 BAR
600	80	95	--
750	95	115	--
900	115	130	--
1000	115	--	--
1200	130	180	200
1400	--	220	240
1800	200	240	320

APPENDIX-

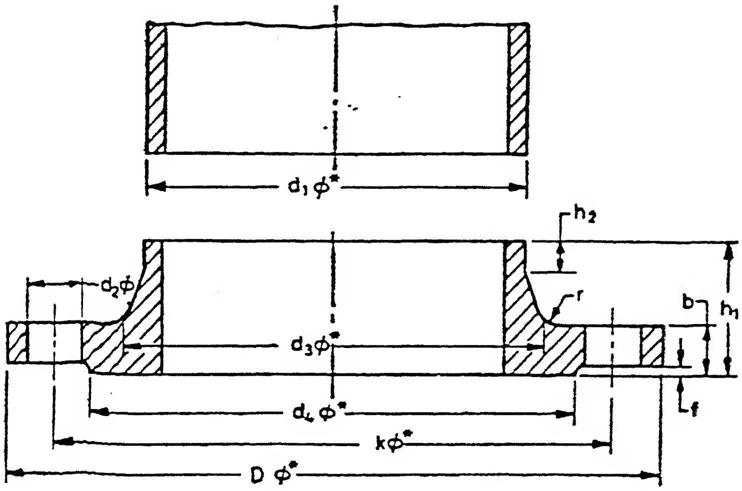
FOR VALVE FLANGE

TABLE 16 WELDING NECK FLANGES

(*Clauses* 4.1 *and* 5.1)

Nominal pressure 1·6 N/mm².

All dimensions in millimetres.



*These dimensions are not to scale.

Nom Size	Pipe o. d. d_1	Flange			Raised Face		Bolting	Drilling			
		D	b	h_1	d_4	f		No.	d_2	k	d_3
10	17·2	90	14	35	40	2	M12	4	14	60	28
15	21·3	95	14	35	45	2	M12	4	14	65	32
20	26·9	105	16	38	58	2	M12	4	14	75	40
25	33·7	115	16	38	68	2	M12	4	14	85	45
32	42·4	140	16	40	78	2	M16	4	18	100	56
40	48·3	150	16	42	88	3	M16	4	18	110	64
50	60·3	165	18	45	102	3	M16	4	18	125	75
65	76·1	185	18	45	122	3	M16	4	18	145	90
80	88·9	200	20	50	138	3	M16	8	18	160	105
100	114·3	220	20	52	158	3	M16	8	18	180	131
125	139·7	250	22	55	188	3	M16	8	18	210	156
150	168·3	285	22	55	212	3	M20	8	22	240	184
175	193·7	315	24	60	242	3	M20	8	22	270	210
200	219·1	340	24	62	268	3	M20	12	22	295	235
250	273	405	26	70	320	3	M24	12	26	355	292
300	323·9	460	28	78	378	4	M24	12	26	410	344
350	355·6	520	30	82	438	4	M24	16	26	470	390
400	406·4	580	32	85	490	4	M27	16	30	525	445
500	508	715	34	90	610	4	M30	20	33	650	548
600	609·6	840	36	95	725	5	M33	20	36	770	652
700	711·2	910	36	100	795	5	M33	24	36	840	755
800	812·8	1025	38	105	900	5	M36	24	39	950	855
900	914·4	1125	40	110	1 000	5	M36	28	39	1 050	955
1 000	1 016	1255	42	120	1 115	5	M39	28	42	1 170	1 058
1 200	1 220	1485	48	130	1 330	5	M45	32	48	1 390	1 262
1 400	1 420	1685	52	145	1 530	5	M45	36	48	1 590	1 465
1 600	1 620	1930	58	160	1 750	5	M52	40	56	1 820	1 668
1 800	1 820	2130	62	170	1 950	5	M52	44	56	2 020	1 870
2 000	2 020	2345	66	180	2 150	5	M56	48	62	2 230	2 072

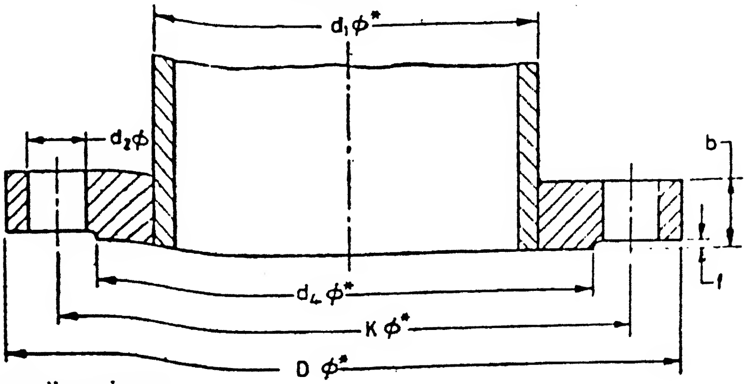
FOR SHAFT FLANGE

TABLE 17 PLATE FLANGES FOR WELDING

(*Clauses* 4.1 *and* 5.1)

Nominal pressure 1·6 N/mm².

All dimensions in millimetres.



*These dimensions are not to scale.

Nom Size	Pipe o. d. d_1	Flange		Raised Face		Bolting	Drilling		
		D	b	d_4	f		No.	d_2	k
10	17·2	90	14	40	2	M12	4	14	60
15	21·3	95	14	45	2	M12	4	14	65
20	26·9	105	16	58	2	M12	4	14	75
25	33·7	115	16	68	2	M12	4	14	85
32	42·4	140	16	78	2	M16	4	18	100
40	48·3	150	16	88	3	M16	4	18	110
50	60·3	165	18	102	3	M16	4	18	125
65	76·1	185	18	122	3	M16	4	18	145
80	88·9	200	20	138	3	M16	8	18	160
100	114·3	220	20	158	3	M16	8	18	180
125	139·7	250	22	188	3	M16	8	18	210
150	168·3	285	22	212	3	M20	8	22	240
175	193·7	315	24	242	3	M20	8	22	270
200	219·1	340	24	268	3	M20	12	22	295
250	273	405	26	320	3	M24	12	26	355
300	323·9	460	28	378	4	M24	12	26	410
350	355·6	520	32	438	4	M24	16	26	470
400	406·4	580	36	490	4	M27	16	30	525
500	508	715	44	610	4	M30	20	33	650
600	609·6	840	52	725	5	M33	20	36	770
700	711·2	910	58	795	5	M33	24	36	810
800	812·8	1 025	64	900	5	M36	24	39	950
900	914·4	1 125	72	1 000	5	M36	28	39	1 050
1 000	1 016	1 255	78	1 115	5	M39	28	42	1 170
1 200	1 220	1 485	94	1 330	5	M45	32	48	1 390